DESIGN STUDY OF A RAREFIED GAS
DYNAMICS AND HEAT TRANSFER CHAMBER

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by

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Lieutenant, United States Navy

Submitted in partial fulfillment of

the requirements for the degree of

MASTER OF SCIENCE

IN

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ABSTRACT

A method for the design of a high vacuum enclosure has been developed.
The method utilizes design procedures recommended by the ASME Boiler and
Pressure Vessel Code where directly applicable. Certain components, which
are not adequately covered by the Code, are designed by elementary stress
analysis. The design method was developed and applied to a specific
design, a rarefied gas dynamics and heat transfer chamber.

ACKNOWLEDGMENTS

The writer wishes to express his sincere appreciation to Professor Virgil M. Faires for the many valuable suggestions he made as consultant for design, and to Professor Paul F. Pucci and Professor Faires for their constant encouragement during the entire thesis project.

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TABLES

1. Introduction: General Design Concepts

In order to simulate high altitude flight conditions encountered by space vehicles re-entering the earth's atmosphere, Professor Paul F. Pucci has proposed a rarefied gas dynamics and heat transfer facility /10/.[1] The facility, shown schematically in Fig. 1, consists of a gas supply, an arc heater which provides the high stagnation enthalpy of the gas, an aero-dynamic nozzle through which the gas is expanded, a test section which is maintained at a high vacuum, an inter-cooler, and a cryopump. The cryopump is essentially a condenser, operating at low enough temperatures (below 40 degrees Kelvin) to condense the gas on its surfaces. The intercooler is used to remove most of the energy from the gas before it enters the cryopump section, in order to minimize the cryopump refrigeration load.

The problem which was posed was the design of a relatively large, ultra-high vacuum vessel and its associated components. The design of the chamber has followed, as closely as possible, the procedure of the ASME Boiler and Pressure Vessel Code /1/, hereinafter referred to as the "Code". Changes in Code procedures were made only when the component design deviated markedly from those components as treated by the Code. Although a specific design has been accomplished in this thesis, the procedure followed here can be readily adapted to the design of any high vacuum system.

The vacuum tank will be built in three flanged sections, each three feet in outside diameter and three feet, between flange centers, in length. The end covers will be dished spherical heads with integral flanges. The interior valving will be housed in separate flanged enclosures. Openings in the shell will be provided for vacuum and cooling connections, access

[1]Numbers appearing in /virgules/ indicate references listed in the Bibliography (p.61).

FIGURE I

RAREFIED GAS DYNAMICS
AND
HEAT TRANSFER FACILITY
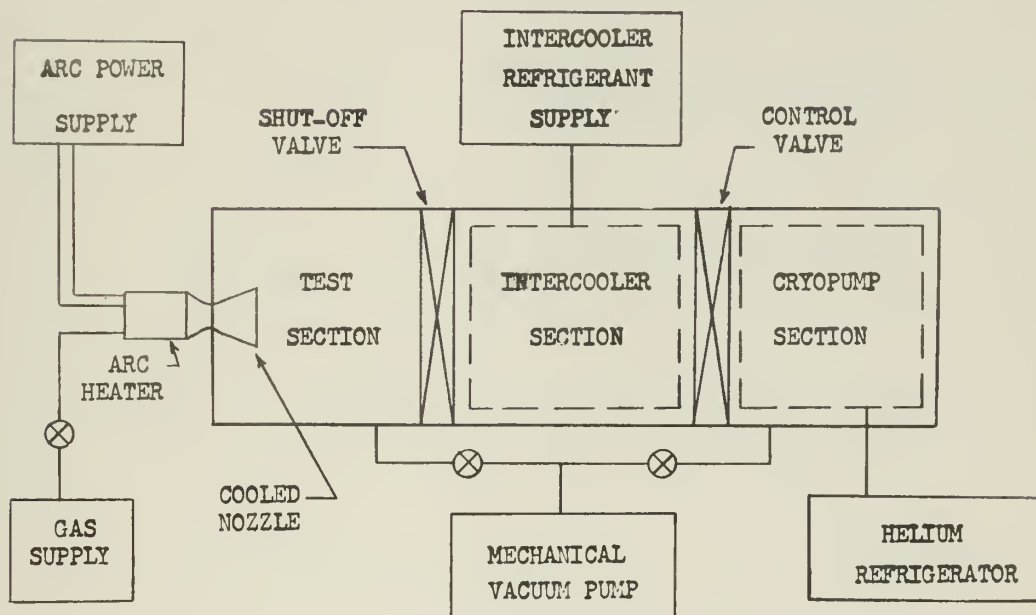
to the test section, access for instrumentation, and observation of the model in the test section. An opening in the front head will be provided for nozzle access. The major components of the chamber are designated in Fig. 2, which is also a top view of the chamber. Overall dimensions are given in Figs. 3a and 3b, side and front views respectively of the chamber. Figure 4 is a dimensioned assembly drawing of the test section.

All required welding shall be done in accordance with the procedures of the Code. No opening in the vessel may penetrate the longitudinal weld joint of the vessel.

FIGURE 2
CHAMBER COMPONENT DESIGNATIONS
(AND TOP VIEW)

OBSERVATION PORT AND
TEST SECTION ACCESS
(11)

INSTRUMENTATION
ACCESS
(8)

SHUT-OFF VALVE
(22) (23)

CONTROL VALVE HOUSING
(17)

STIFFENING RING

REAR HEAD
(13)

SECTION JOINING FLANGES
(7)

NAZZLE ACCESS
(14) 15

FRONT HEAD
(13)

← TEST SECTION →
(14)

← INTERCOOLER SECTION →

← CRYOPUMP SECTION →
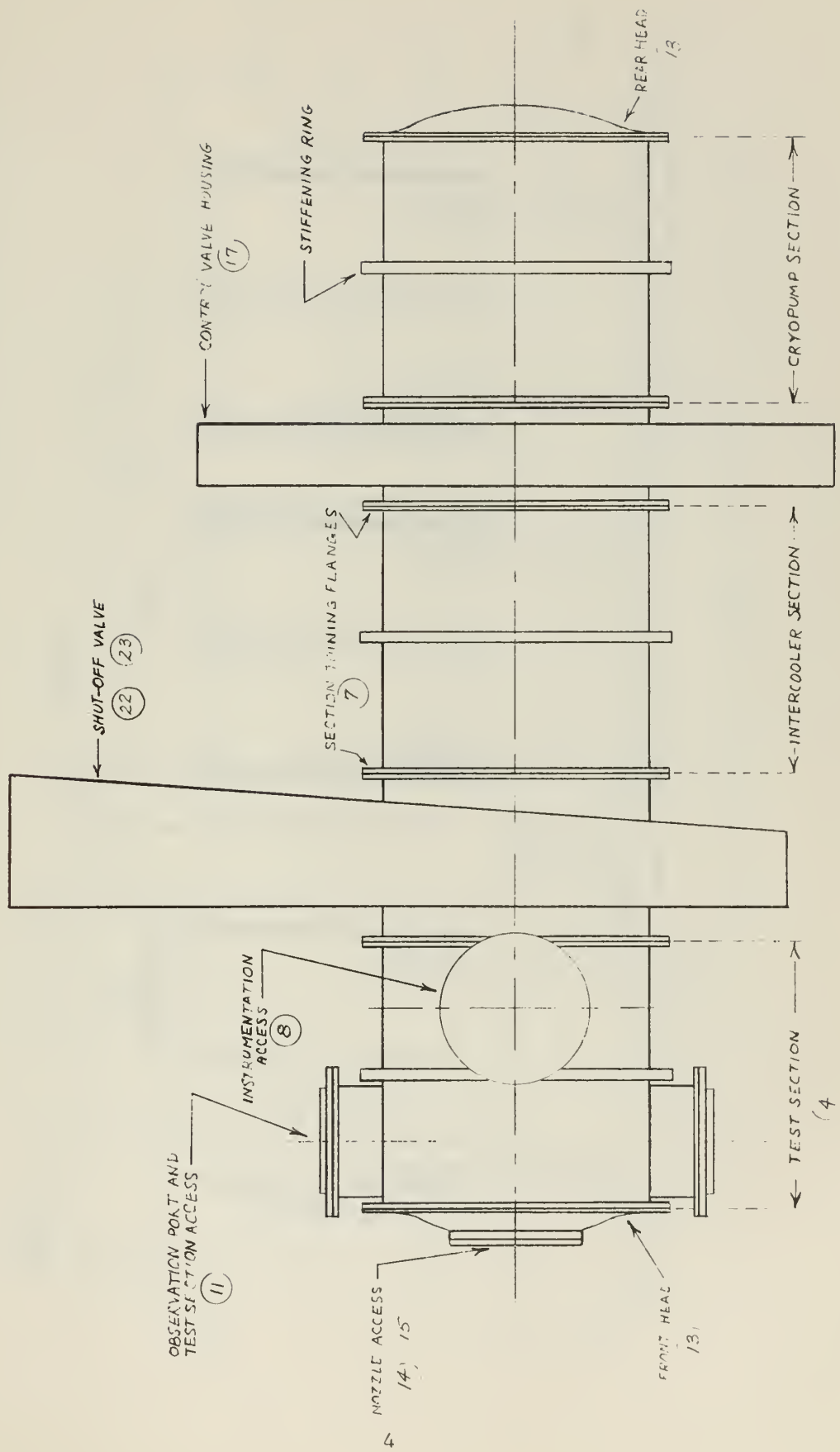
NOTE : CIRCLED NUMBERS IN ALL DRAWINGS REFER TO FIGURE NUMBERS

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FIGURE 3a

CHAMBER SIDE VIEW

FIGURE 3 b

CHAMBER FRONT VIEW

FIGURE 4

TEST SECTION ASSEMBLY

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2. Material Requirements.

When a high vacuum is pumped, surfaces of the vacuum enclosure release quantities of gas which inhibit the attainment of an ultra-high vacuum. In addition to those gases released due to dissolution from the solid material and those bound to the surface of the solid by physical adsorption, large quantities of gases are released from any interior surface oxides present. Gases may also enter the vacuum enclosure by permeation of the metal walls. The phenomenon of gas release into a vacuum enclosure 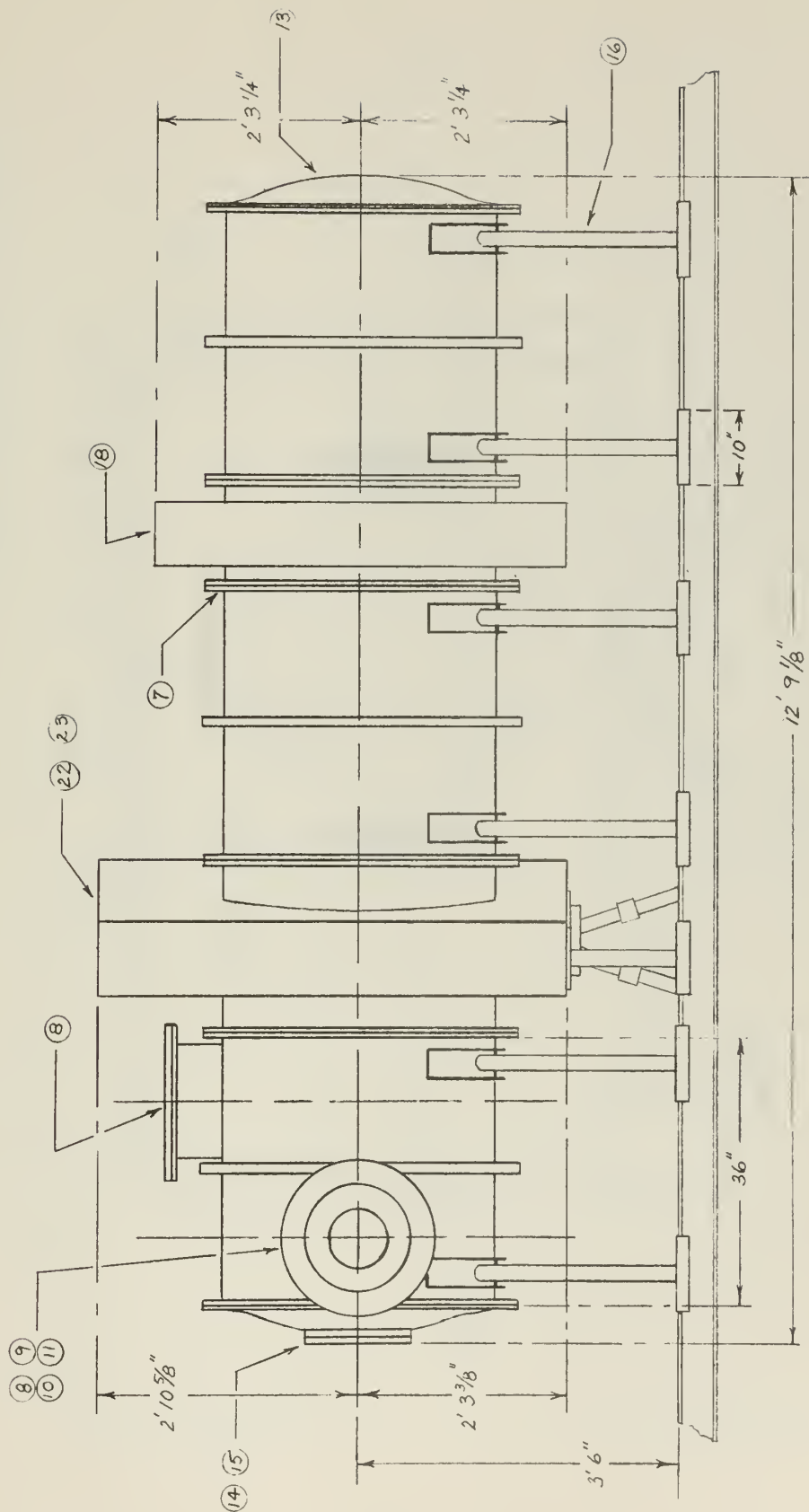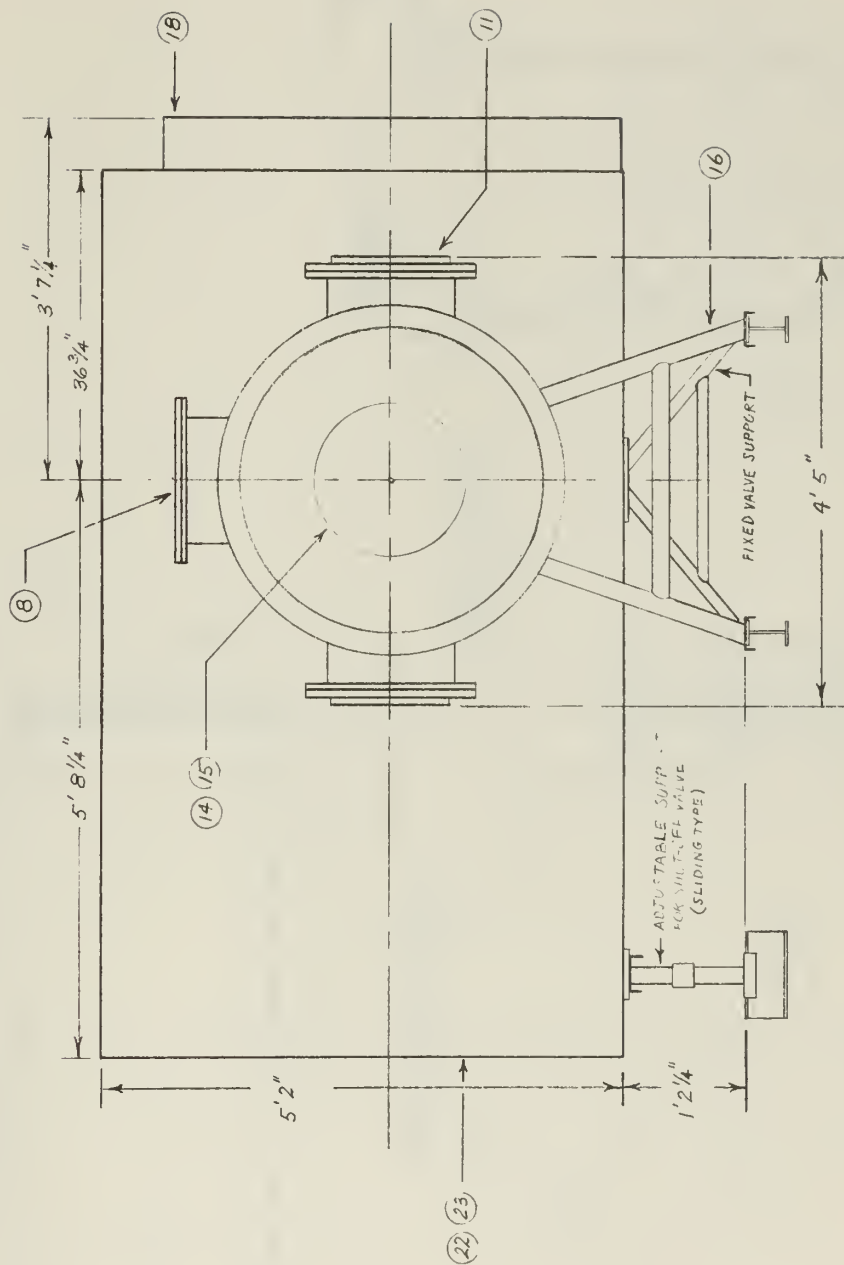is sometimes referred to as outgassing /2/, /3/, /4/, /5/. The rate of evolution of gas from the surface is a function of time, and appears to vary exponentially /3/.

During pumping of a closed system, pressure decreases continually, and the rate of decrease of pressure also decreases. The "ultimate" pressure for any system is reached when equilibrium exists between the amount of gas released into the system and the amount which is pumped /4/, /6/.

The attainment of ultra-high vacua by ordinary vacuum pumping takes considerable time. The objectives of cryopumping are (1) to appreciably reduce the time required to obtain a given vacuum, (2) to enable a higher vacuum to be obtained than that which is possible by ordinary vacuum pumping, and (3) to maintain the vacuum in a chamber into which a gas is purposely introduced.

Hydrogen is a highly objectionable gas in a cryopump system because of its high thermal conductivity and its resistance to condensation on cold surfaces. Another objectionable "gas" in a high vacuum cryopump system is water vapor, which when introduced into the system, acts as a ready source of hydrogen gas.

Types 304 and 347 stainless steels, which are endowed with good tensile

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properties and impact resistance at both ordinary and cryogenic temperatures, are used extensively in cryogenic equipment. These steels have relatively low thermal conductivities. When one considers permeation of hydrogen through the metal walls of high vacuum systems, it may be said that the use of chrome steels in lieu of plain carbon steels for these walls, in effect, reduces the permeation by several factors of ten. Type 347 stainless steel has better stability under the heat of welding than does Type 304.

In considering all the factors involved in selection of material, one is led to the conclusion that the use of Type 347 stainless steel is called for in this design. Some properties of this material (in a heat treated condition) as required by Section II of the Code are listed in Table I.

TABLE I   REQUIREMENTS FOR TYPE 347 STAINLESS STEEL

Min. Tensile Strength, psi	75,000
Min. Yield Strength, psi (0.20% offset)	30,000
Min. Elongation in Two Inches, %	40.0
Max. Hardness Number	202 Brinell or 88 Rockwell B
Chemical Requirements	0.08(max)C; 2.00(max)Mn; 0.045(max)P; 0.03(max)S; 1.00(max)Si; 17-19 Cr; 9-13Ni; Cb+Ta, 10xC(min), 1.10(max)
Welding Electrodes	E347 (18-8Cb or 19-9Cb)

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3. The Basic Vacuum Vessel and Stiffening Rings

    a. The rules for design of a cylindrical shell under external pressure are found in Section 8, paragraph UG-28 of the Code. The known dimensions for each section of the tank are the outside diameter, D_o, and the total length of the section. Here it is assumed that a circumferential stiffening ring will be installed at the middle cross-sectional plane of each tank section. The flanges on each end of the tank sections are considered to be stiffening rings. The design length, L, therefore is the greatest center-to-center distance between any two adjacent stiffening rings.

    The symbols defined below are used in the formulae of this section:

        t = minimum required thickness of cylindrical shell plates.
           (No corrosion allowance is used since material is corrosion
           resistant chromium-nickel steel).

        L = design length of a vessel section, taken as the greatest
           center-to-center distance between any two adjacent stiffen-
           ing rings. Here, L = 18 inches.

        D_o = outside diameter of the cylindrical shell course under
           consideration. Here, D_o = 36 inches.

        P_a = maximum allowable working pressure, psi, computed by:

$$P_a = \frac{B}{D_o/t} ; \text{ where B is a constant obtained (1)}^{[1]}$$
$$\text{from a plot in the Code.}$$

        P_c = collapsing pressure obtained by rational means, psi.

        P_e = external design pressure, psi. Here, P_e = 15 psi.

    Several values of thickness were assumed before closure was achieved between P_a and P_e . Using a value of t = 0.1 inch,

        L/D_o = 18/36 = 0.5,

        D_o/t = 36/0.1 = 360.

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[1]Numbers appearing in parentheses at right of page are formula numbers.

Entering Fig. UA-28.2, Section 8 of the Code obtain the following value

of B:

B = 6600 (corresponding to maximum temperature line of 100F).

From formula (1), the maximum allowable working pressure is:

P_a = 18.3 psi.

Further calculations for refinement of the required thickness are un-

necessary. We will use, t = 0.1 in., as the required thickness by the Code.

The collapsing pressure obtained by rational means from paragraph 460,

Faires /7/, is P_c = 62.3 psi for this thickness.

An actual design thickness of shell is chosen to be:

t_{actual} = 1/4 inch.

The collapsing pressure for this thickness, from Faires /7/, is P_c = 633

psi. The reasons for choosing a thickness two and one half times larger

than that required by the Code are:

(1) Deflections of the tank are critical when considering mating

flanges for "perfect" vacuum seal,

(2) Certain internal components must be supported,

(3) Excess thickness over that required by the Code serves to

act as reinforcement of shell openings.

Using a design thickness of 1/4-inch, the maximum allowable working

pressure by the Code is, P_a = 82 psi. From the thin shell formula /7/,

using the minimum yield strength, s = 30,000 psi, for stress at the elastic

limit, and a welding efficiency of 85 percent, the design factor, N, is more

than 23.

From Section 8, paragraph UG-80 of the Code, it is determined that the

maximum deviation (plus or minus) from the true circular form shall not

exceed 0.115 inches measured radially in a chord length of 7.2 inches.

b. The design of stiffening rings for cylindrical shells under exter-
nal pressure is covered in Section 8, paragraph UG-29 of the Code. In
addition to the symbols previously defined in this section, the following
symbols are used in design of stiffening rings:

I_s = required moment of inertia of stiffening ring about its
    neutral axis, parallel to the axis of the shell, inches
    fourth.

A_s = cross-sectional area of stiffening ring, sq. in.

A = a factor determined by type material, design temperature
    limit, and a factor, B

B = a factor determined by:

$$B = \frac{P_e D_o}{t + A_s/L_s} \tag{2}$$

L_s = design length of the vessel as previously determined.
    Here, L_s = 18 inches.

The required moment of inertia of a circumferential stiffening ring
shall not be less than that determined by:

$$I_s = \frac{(D_o)^2 L_s A}{14} (t + A_s/L_s) \tag{3}$$

After several trials for closing the values of I_{actual} and I_s, a standard
3-in., 4.1-lb-per-foot channel is selected for the job. B is calculated
from formula (2) and A is obtained from Fig. UA-28.2, Section 8 of the Code.
The required moment of inertia by eq. (3) is:

$$I_s = 0.0632 \text{ inches fourth.}$$

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The actual moment of inertia of the channel is:

$$I_{actual} = 1.60 \text{ inches fourth.}$$

With a design factor of five, a rational approach /7/ yields a required moment of inertia of:

$$I_{rational} = 0.079 \text{ inches fourth.}$$

The stiffening rings shall be welded around the outside circumference of the vessel. Intermittent welding may be used, with weld length on each side of the ring equal to not less than one half of the circumference.

The reason for choosing a stiffening ring with an actual moment of inertia so much larger than that which was required, was to allow use of an available standard channel iron.

4. Openings and Reinforcement in the Vacuum Enclosure

The following is a list of all openings in the shell and heads of the vacuum enclosure:

        a. Test Section Accesses (two required)

        b. Instrumentation Access

        c. Nozzle Access

        d. Cooling and Vacuum Connections (eight required)

The test section accesses will be located forward of the transverse center line, with center on the horizontal center line of the test section. They will be fabricated of 5/16-inch material and will extend, on center line, a distance of six inches from tank wall to inside of flange. Figure 11 is an assembly drawing of the test section access-observation port configuration. The instrumentation access will be similar to the test section accesses except that the instrumentation access flange will be blanked-off with a 7/8-inch blank flange in place of the observation port supporting bracket. The blank flange may be drilled for installation of instrumentation.

The nozzle assembly access is shown in detail in Figs. 14 and 15.

There will be vacuum connections in the test section and in the cryopump section, which will connect to standard two-inch pipe. There will be two liquid air connections in the intercooler section and two in the cryopump section. There will be two helium connections in the cryopump section. The latter six connections will be flanged directly to two-inch jacket piping on the external coolant lines. Therefore, all eight cooling and vacuum connections are identical in design. All will be made of Type 347 stainless steel, two-inch, standard (schedule 40) pipe with wall thickness of 0.154 in.,

projecting four inches, on center line, from the tank wall to the inside of the flange. The flange is illustrated in Fig. 12.

From para. UG-37, (c) of the Code, the requirement for a reinforcement cross-sectional area around an opening in the vessel is:

$$A = 0.50 \ d \ t_r \ , \text{ where } d = \text{diameter in plane of finished opening,}$$
$$t_r = \text{wall thickness required by the Code}$$

a. For the test section and instrumentation accesses, $d = 15$ in., $t_r = 0.10$ in. Therefore the required reinforcement area is: $A = 0.75$ sq. in.

b. For the nozzle access in the front head, $d = 12.5$ in., $t_r = 0.292$ in. Therefore the required reinforcement area is: $A = 1.825$ sq. in.

c. All openings for cooling and vacuum connections, due to their size, are specifically exempted reinforcement by the provisions of para. UG-36, (c), (3) of the Code.

All metal of the walls of thickness greater than that required by the Code is included in reinforcement.

a. For the instrumentation and test section accesses, the excess metal in the walls available for reinforcement by para. UG-40 of the Code is:

$$A_1 = (t - t_r) \ d, \text{ where } t = \text{actual thickness.}$$

Since $t = 0.250$ in., $A_1 = 2.25$ sq. in., which is greater than the area required for reinforcement.

b. For the nozzle access opening, $t = 0.500$ in., $A_1 = 2.60$ sq. in., which is greater than that required.

It is seen from the above calculations that additional reinforcement of the openings in the vacuum enclosure is unnecessary.

5. General Flange Design.

    Design of bolted flange connections is covered in Appendix II to
Section 8 of the Code. Paragraph UA-55 specifically covers flanges subject-
ed to external pressure. The symbols defined below are used in the formulae
of this section:

   A = outside diameter of flange, inches

   A_b = actual total stress area of bolts, sq. in.

   A_{m2} = total stress area of bolts required for gasket seating,
        sq. in., $A_{m2} = W_{m2}/S_a$

   B = inside diameter of flange, in.

   C = bolt circle diameter, in.

   d = factor for integral type flanges, $d = \dfrac{U\, h_o\, (g_o)^2}{V}$

   e = factor for integral type flanges, $e = F/h_o$

   F = factor for integral type flanges (Fig. UA-51.2 of Code)

   f = hub stress correction factor for integral flanges
        (Fig. UA-51.6 of Code)

   G_m = mean diameter of gasket load reaction, in.

   g_o = thickness of hub at small end, in.

   g_1 = thickness of hub at back of flange including length of
        weld leg, in.

   H = total hydrostatic end force, lbs., $H = 0.785 P_e\, (G_m)^2$

   H_D = hydrostatic end force on area inside of flange, lbs.,
        $H_D = 0.785 P_e\, (B)^2$

   H_T = difference between H and H_D, lbs., $H_T = H - H_D$

   h = hub length, in.

   h_D = radial distance from bolt circle to circle on which
        H_D acts, in., $h_D = R + 0.5\, g_1$

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h_G = radial distance from gasket load reaction to bolt circle, in., $h_G = (C - G_m)/2$

h_o = flange factor, $h_o = (B\,g_o)^{1/2}$

h_T = radial distance from bolt circle to circle on which H_T acts, in., $h_T = (R + g_1 + h_G)/2$

K = ratio of outside to inside diameters, $K = A/B$

L = factor, $L = \dfrac{(t\,e + 1)}{T} + \dfrac{(t)^3}{d}$

M_o = total moment acting on flange for gasket seating or operating conditions, whichever is greater, in.-lbs.

P_e = external design pressure, psi

Q = force on flange due to compression of "O" ring gaskets, lbs.-per-inch of circumference, per gasket

R = radial distance from bolt circle to intersection of hub and back of flange, in., $R = 1/2(C - B) - g_1$

S_a = allowable bolt stress at atmospheric temperature, psi

S_b = allowable bolt stress at design temperature, psi, $S_b = S_a$ for this design

S_f = allowable design stress for flange material at design temperature (use smaller value obtained for forgings, castings, or plate steel), psi

S_n = allowable design stress for material of vessel or pipe wall at design temperature (use smaller value obtained for seamless or welded), psi

S_H = calculated longitudinal stress in hub, psi

S_R = calculated radial stress in flange, psi

S_T = calculated tangential stress in flange, psi

T = factor involving K (Fig. UA-51.1 of Code)

t = effective flange thickness, in.

U = factor involving K (Fig. UA-51.1 of Code)

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V = factor for integral type flanges (Fig. UA-51.3 of Code)

W = flange design bolt load for gasket seating, lbs.,
   $W = 1/2(A_{m2} + A_b) S_a$

W_{m2} = minimum required bolt load for gasket seating, lbs.,
   $W_{m2} = 3.14 \ G_m \ Q$

Y = factor involving K (Fig. UA-51.1 of Code)

Z = factor involving K (Fig. UA-51.1 of Code)

   A summary of the values of some of the constants used in substantiating

the design of the various flanges is given in Table II. Each major flange

combines the use of two "O" rings and a vacuum pump-out to effect a good

seal. The vacuum pump-out consists of a groove of given dimensions and a

1/8-inch hole, drilled as shown on the individual sketches of the flanges,

to which a vacuum connection is made. A cross-sectional view of a vacuum

pump-out groove is shown in Fig. 5.

   The effective thickness of flange used in the calculations is the total

thickness of the flange minus the depth of cut for the "O" ring grooves and

vacuum pump-out groove. A safe design is thus provided inasmuch as the ad-

ditional metal around the grooves has not been considered in the strength

calculations. Table III and its accompanying sketch give dimensions of the

"O" ring grooves and the cross-sectional diameter of the "O" ring for each

type of fitting. An end view of a typical flange is shown in quarter-sec-

tion in Fig. 6 and dimensions for all flanges are given in Table IV.

   Parts A, B, C, D and E of this section give a summary of the calcula-

tions made to substantiate the design of the flanges and fittings. A com-

plete set of calculations for the design of one type of flange is given in

Appendix I. The design of these flanges followed the Code procedure as

closely as possible. Using data received from a manufacturer of "O" rings

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/8/ and that obtained from "O" ring catalogs, changes were made in Code calculations as follows:

(a) The determination of W_{m2}, minimum required bolt load for gasket seating in pounds, was made by using one and one third times the manufacturer's recommended gasket force of 15 pounds per lineal inch of circumference per "O" ring, times the mean circumference of gasket reaction. This was done in lieu of the method of the Code.

The reasons for deviation from the method of the Code were:

(a) The Code considers only single gasketing of a flange.

(b) The only ring joint gasket considered by the Code is a metal gasket.
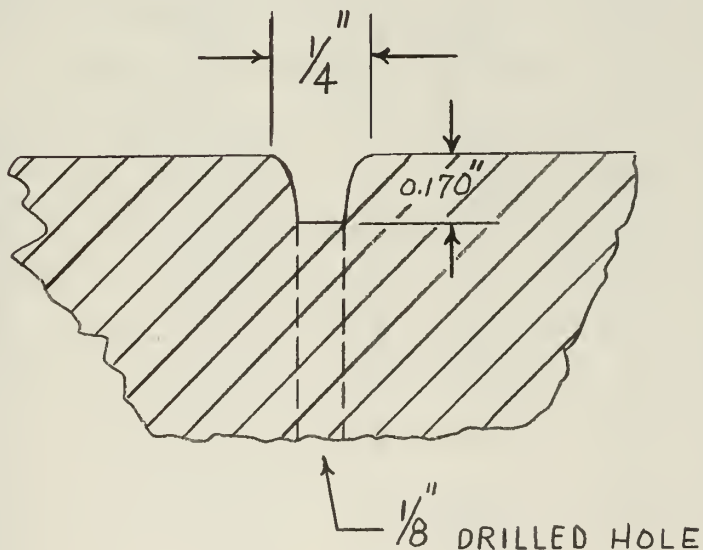
Figure 5

Vacuum Pump-Out Groove

TABLE II

CONSTANTS USED IN FLANGE DESIGN

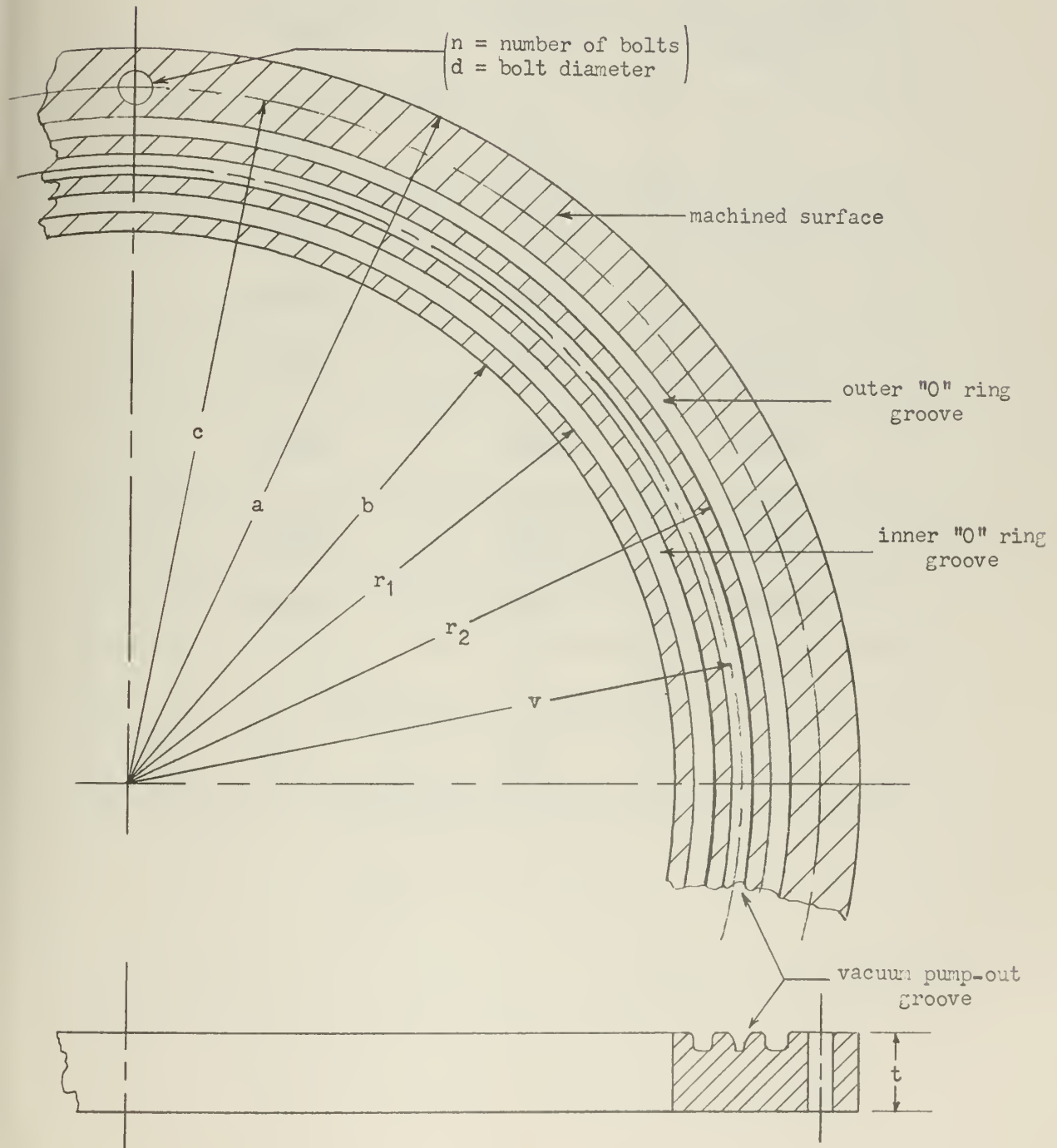|	Section Joining Flange	External Piping Flange	Observation Port Locking Ring	Test Section Access Flange
A	41.375	7.125	14.120	20.026
A_b	4.3968	0.6184	1.2368	1.2368
A_{m2}	0.316	0.0336	0.0855	0.1405
B	36.00	2.375	8.00	15.000
C	40.125	6.180	13.040	19.020
F	0.902	0.855	—	0.392
f	4.35	2.70	—	3.80
G_m	37.745	4.011	10.235	16.745
g_o	0.250	0.154	—	0.3125
g_1	0.605	0.372	—	0.7545
h	0.355	0.218	—	0.4420
h_o	3.00	0.6045	—	2.165
K	1.15	3.00	1.767	1.335
L	0.660	2.07	—	0.7665
P_e	15.0	15.0	15.0	15.0
Q	20	20	20	20
S_a	15,000	15,000	15,000	15,000
S_f	17,500	17,500	17,500	17,500
S_n	16,000	16,000	—	16,000
T	1.85	1.20	—	1.70
t	0.530	0.455	0.392	0.530
U	17.0	2.20	—	8.00
V	0.41	0.235	—	0.335
W	35,346	4,890	9,920	10,330
W_{m2}	4,740	503.8	1285.5	2103.2
Y	15.0	2.00	3.60	7.00
Z	8.0	1.30	—	3.50

TABLE III

"O" RING GROOVE DIMENSIONS
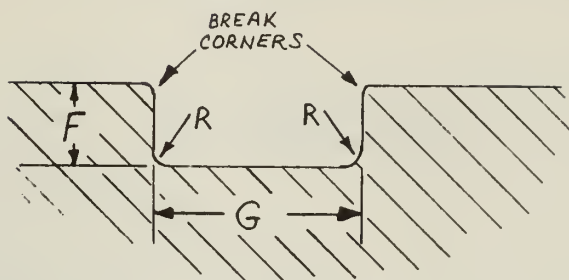
	Section Joining Flange FIG. 7	External Piping Flange FIG. 12	Obs. Port Locking Ring FIG. 9	Obs. Port Support Bracket FIG. 10	Inst. and Test Section Accesses FIG. 8	Nozzle Access Ring FIG. 14
Diameter of "O" Ring Cross Section, W Tol.= ± .005	.210	.210	.210	.210	.210	.210
Inside Diameter of Grooves Tol.= + .005	(1)36.500 (2)38.563	(1) 2.865 (2) 4.737	8.755	(1) 8.755 (2)11.505	(1)15.500 (2)17.570	(1)13.005 (2)15.005
Groove Depth, F Tol. = ± .005	.170	.170	.170	.170	.170	.170
Groove Length, G Tol. = ± .005	.282	.282	.282	.282	.282	.282
Radius R Tol. = ± .005	.025	.025	.025	.025	.025	.025

FIGURE 6

FLANGE END VIEW-QUARTER SECTION

NOTES:

1. See Table IV for flange dimensions

TABLE IV

FLANGE DIMENSIONS

	Section Joining Flange	External Piping Flange	Instrumentation and Test Section Accesses
a	20.683	3.564	10.013
b	18.000	1.188	7.500
c	20.063	3.089	9.510
r_1	18.250	1.433	7.750
r_2	19.282	2.369	8.785
t	0.750	0.625	0.750
v	18.907	2.041	8.410
n	32	8	16
d	1/2	3/8	3/8

A. Tank Section Joining Flange

This flange is shown in half-section in Fig. 7. Substitution of the values obtained from Table II into the formulae of Appendix II to Section 8 of the Code yielded the following results:

$$H_D = 0.785 \; P_e \; (B)^2 = 0.785 \times 15 \times 36^2 = 15,260 \; \text{lbs.,}$$

$$H = 0.785 \; P_e \; (G_m)^2 = 0.785 \times 15 \times 37.745^2 = 15,780 \; \text{lbs.,}$$

$$H_T = H - H_D = 520 \; \text{lbs.}$$

The moment arms were solved for as listed in the table of symbols.

(a) From paragraph UA-55 of the Code, for operating conditions:

$$M_o = H_D \; (h_D - h_G) + H_T \; (h_T - h_G) = 8,940 \; \text{in.-lbs.}$$

(b) From paragraph UA-55 of the Code, for gasket seating:

$$M_o = W \; h_G = 42,062 \; \text{in.-lbs.}$$

It is seen that the moment due to gasket seating determines the flange design. Substituting the values from Table II into the stress formulae of paragraph UA-51 of the Code, the following stress values are calculated:

$$S_H = \frac{f \; M_o}{L \; B(g_1)^2} = 21,000 \; \text{psi,}$$

$$S_R = \frac{(1.33 \; t \; e + 1) \; M_o}{L \; B \; (t)^2} = 6,500 \; \text{psi,}$$

$$S_T = \frac{Y \; M_o}{(t)^2 B} - Z \; S_R = 200 \; \text{psi.}$$

The requirements for strength as found in paragraph UA-52 of the Code compare with the calculated values as follows:

(1) $S_H < 1.5 \; S_n$; $21,000 < 1.5 \times 16,000.$

(2) $S_R < S_f$; $6,500 < 17,500$,

$S_T < S_f$; $200 < 17,500$.

(3) $1/2 \, (S_H + S_R) < S_f$; $13,750 < 17,500$,

$1/2 \, (S_H + S_T) < S_f$; $10,600 < 17,500$.

(4) Shearing stress on weld area must be less than $0.8 \, S_n$.

Area of weld in shear $= 3.14 \, D_o$ x Length of weld,

$A = 3.14 \times 36 \times 0.853$.

Force to shear weld $= W_{m2} = 4740$ lbs.

$S_s = 49.1$ psi.

The design of this flange is substantiated. A mating flange of the same thickness, without "O" ring grooves, is of conservative design. The mating flange of the end enclosure is designed integrally with the head (See Section 6, page 37).

Appendix I contains a complete set of computations for this flange.

FIGURE 7
SECTION JOINING FLANGE
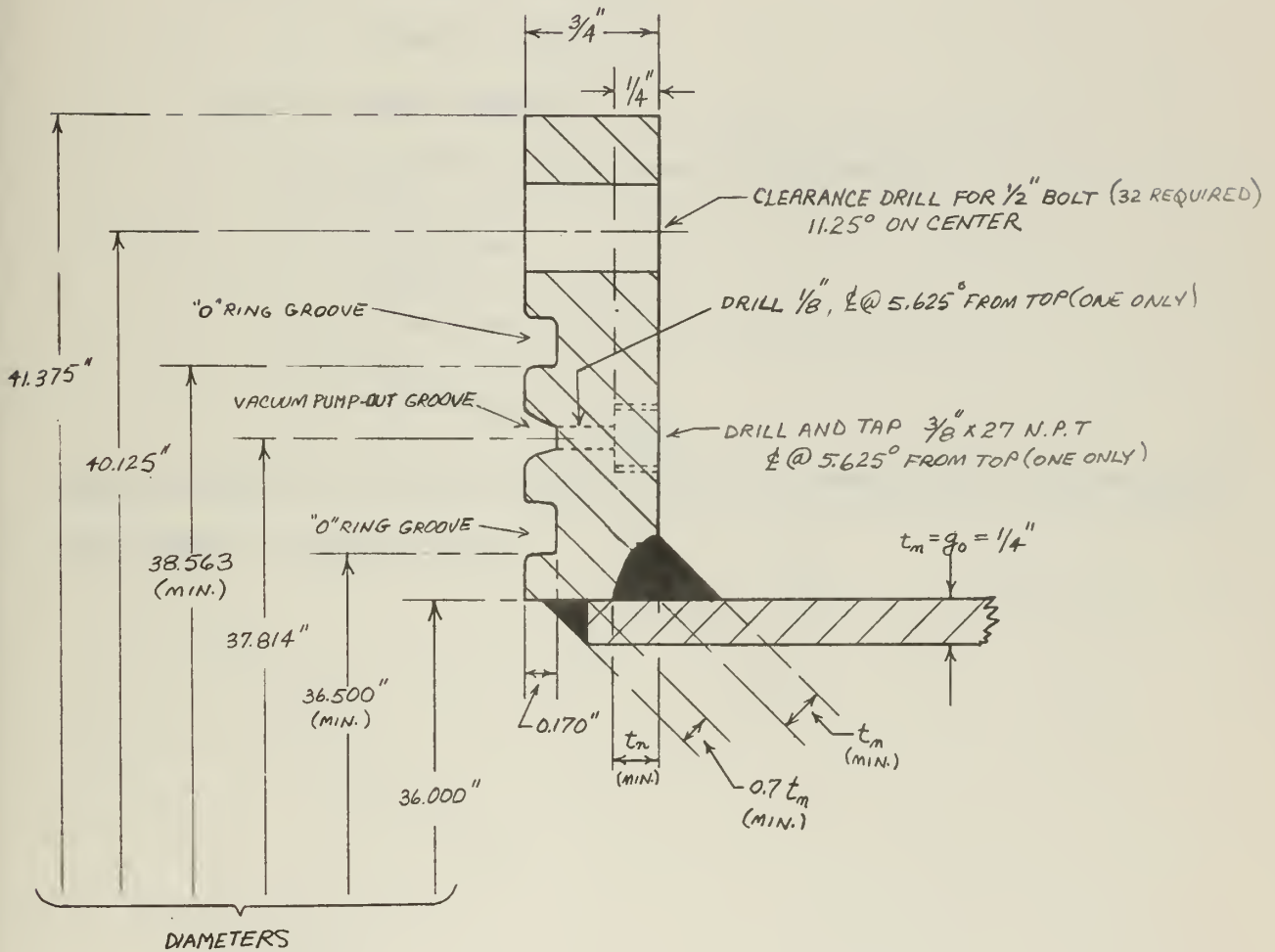
NOTES:
1. Vacuum Pump-Out Groove dimensions given in Figure 5
2. "O" Ring Groove dimensions shown in Table III
3. Tolerance on all min. dimensions is + 0.005 in.
4. Tolerances not shown are ± 0.005 in.
5. See Figure 6 for end view

B. Instrumentation and Test Section Access Flanges

This flange is shown in half-section in Fig. 8. Using the values obtained from Table II and following the same procedure as in Part A of this section, the results listed below were obtained:

(a) For operating conditions:

M_o = 1,598 in.-lbs.

(b) For gasket seating:

M_o = 11,750 in.-lbs. (controlling design),
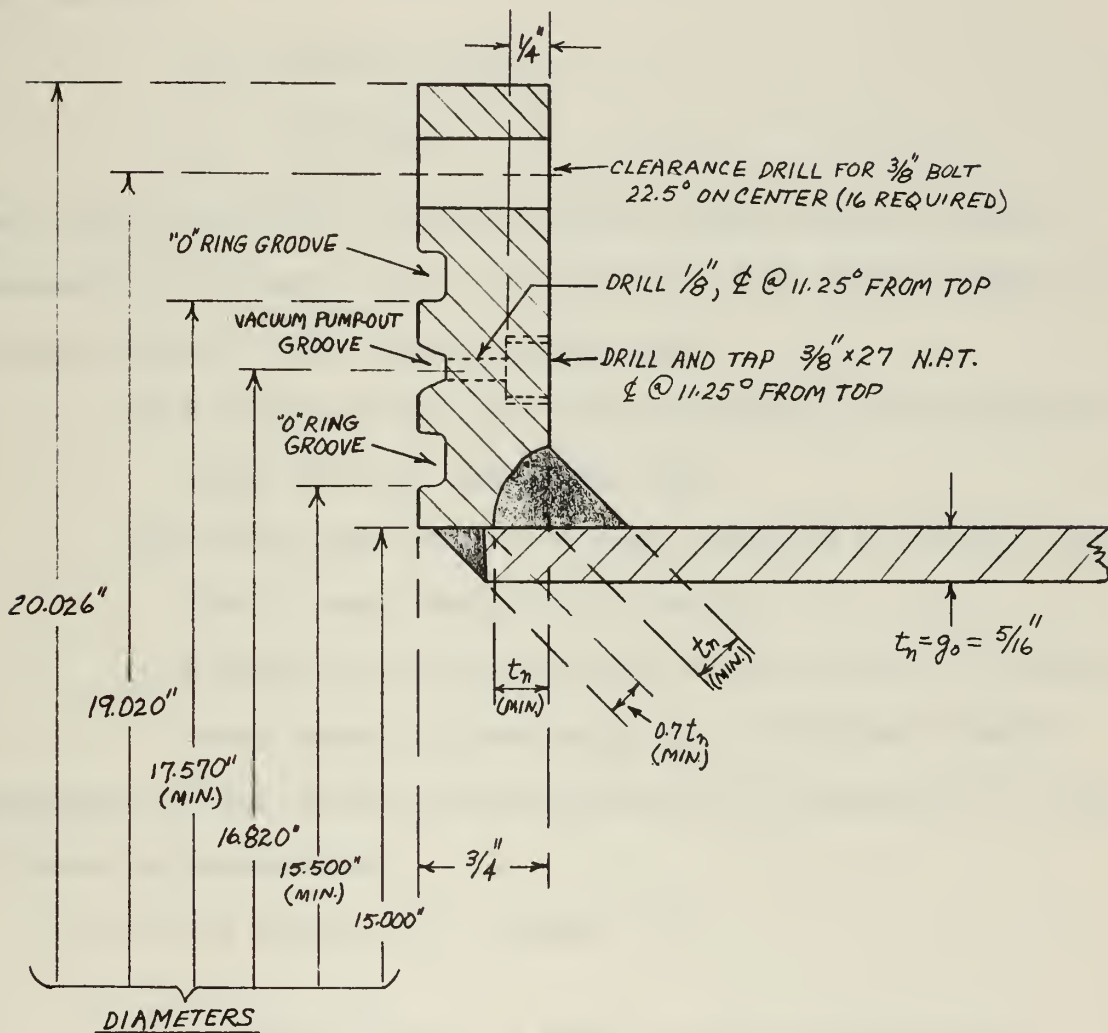
S_H = 6,840 psi,

S_R = 4,000 psi,

S_T = 2,300 psi.

Shearing stress on the weld, S_s = 41.9 psi, is conservative. The design of this flange is substantiated.

FIGURE 8
INSTRUMENTATION AND TEST SECTION
ACCESS FLANGES

NOTES:
1. Vacuum Pump-Out Groove dimensions given in Figure 5
2. "O" Ring Groove dimensions shown in Table III
3. Tolerance on all min. dimensions is + 0.005 in.
4. Tolerances not shown are ± 0.005 in.
5. See Figure 6 for end view

C. Observation Port Locking Ring

The locking ring may be considered a loose type ring flange. This flange is illustrated in Fig. 9. Using the data from Table II, in accordance with paragraph UA-51, (2) of the Code, the following results were obtained:

$$M_o = 13,913 \text{ in.-lbs},$$

$$S_T = 40,800 \text{ psi.}$$

Due to this high value of stress, a check was made using the theory as developed by Timoshenko /9/ for circular plates with circular holes cut in their centers. Three assumptions were made:

(a) Consider all the force from both gaskets to be distributed along the inner edge of the ring,

(b) Consider the plate to be simply supported at the bolt circle with no moment applied by the bolts,

(c) Consider the simple support at the bolt circle to be distributed instead of considering the 16 individual supports.

Assumptions (a) and (b) make the design extremely conservative; (c) tends to reduce the conservatism.

From formula (75), p. 67, Timoshenko /9/,

$$\sigma_{max} = (kP)/(h)^2$$ where; k depends on the ratio of outside
to inside radii,
P is the total load applied to the plate,
and
h is the plate thickness.

The load, P, is calculated using the gasket reaction force of 20 pounds per lineal inch of circumference per gasket.

$P = 20 \ (\ 3.14 \ x \ 8.86 + 3.14 \ x \ 11.61 \ ) = 1288 \ \text{lbs.}$,
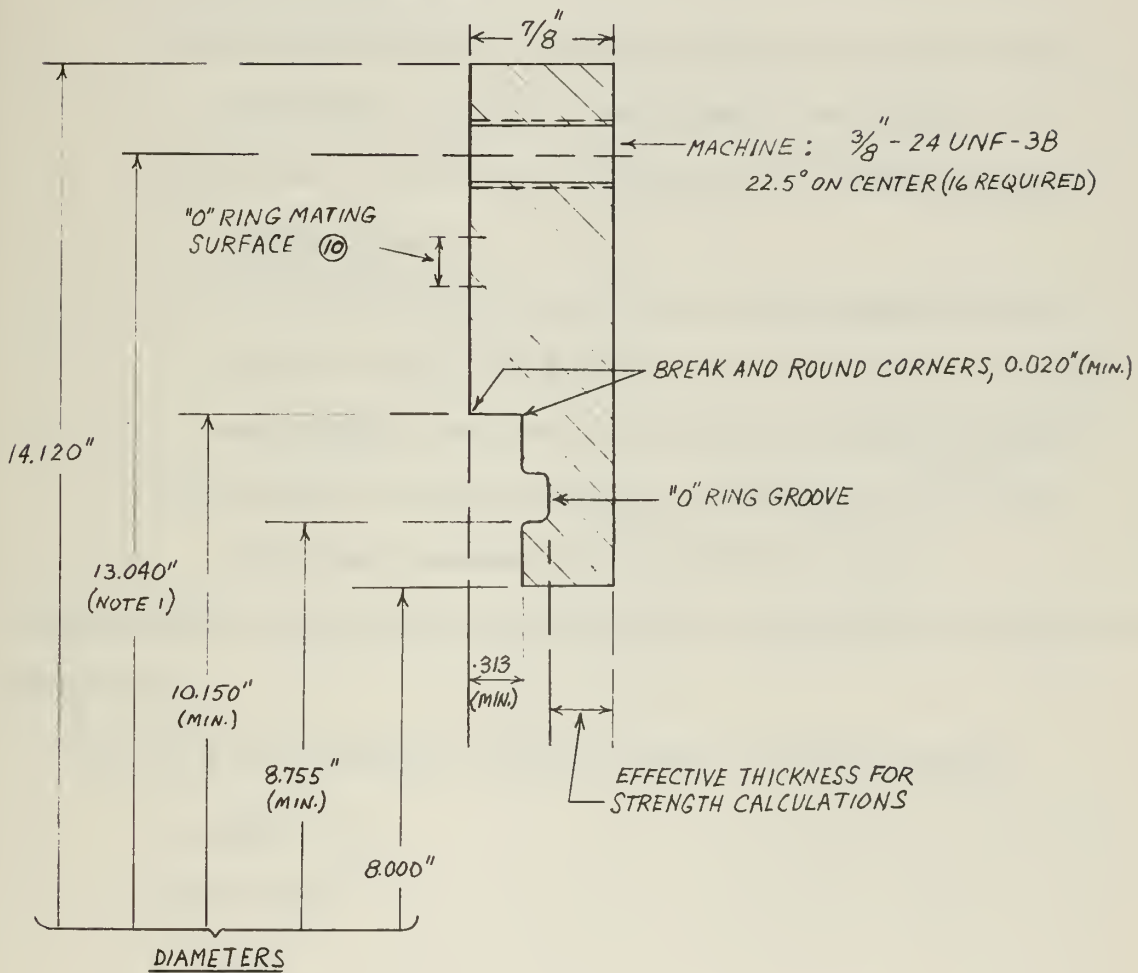
$A/B = 1.595$, use 1.60,

$k = 1.304$ by interpolation in Table 3, p. 68, Timoshenko.

$$\sigma_{max} = 10,920 \ \text{psi.}$$

Because of the ultra-conservative assumptions and the use of the minimum flange thickness in the calculations, the minimum design factor as calculated by classical means is, $N = 6.40$. The design as shown in Fig. 9 is conservative.

FIGURE 9

OBSERVATION PORT
LOCKING RING

D. Observation Port Supporting Bracket

The observation port supporting bracket is designed in accordance
with the theory as developed by Timoshenko, without recourse to the method
of the Code. A sketch of the bracket is shown in Fig. 10. The test section
access-observation port assembly is shown in Fig. 11.

Four assumptions were made:

(a) Consider all the force from both bottom gaskets to be
distributed along the inner edge of the plate,

(b) Assume no force due to bolting of the observation port
locking ring,

(c) Assume that the force of the two upper gaskets acts at
the bolt circle and does not tend to decrease the moment,
and consider that the bolt support is a simple support,

(d) Consider the simple support at the bolt circle to be
distributed instead of 16 individual supports.

Assumptions (a), (b) and (c) were conservative; (d) tended to reduce the
conservatism.

Following the procedure outlined in part C of this section:
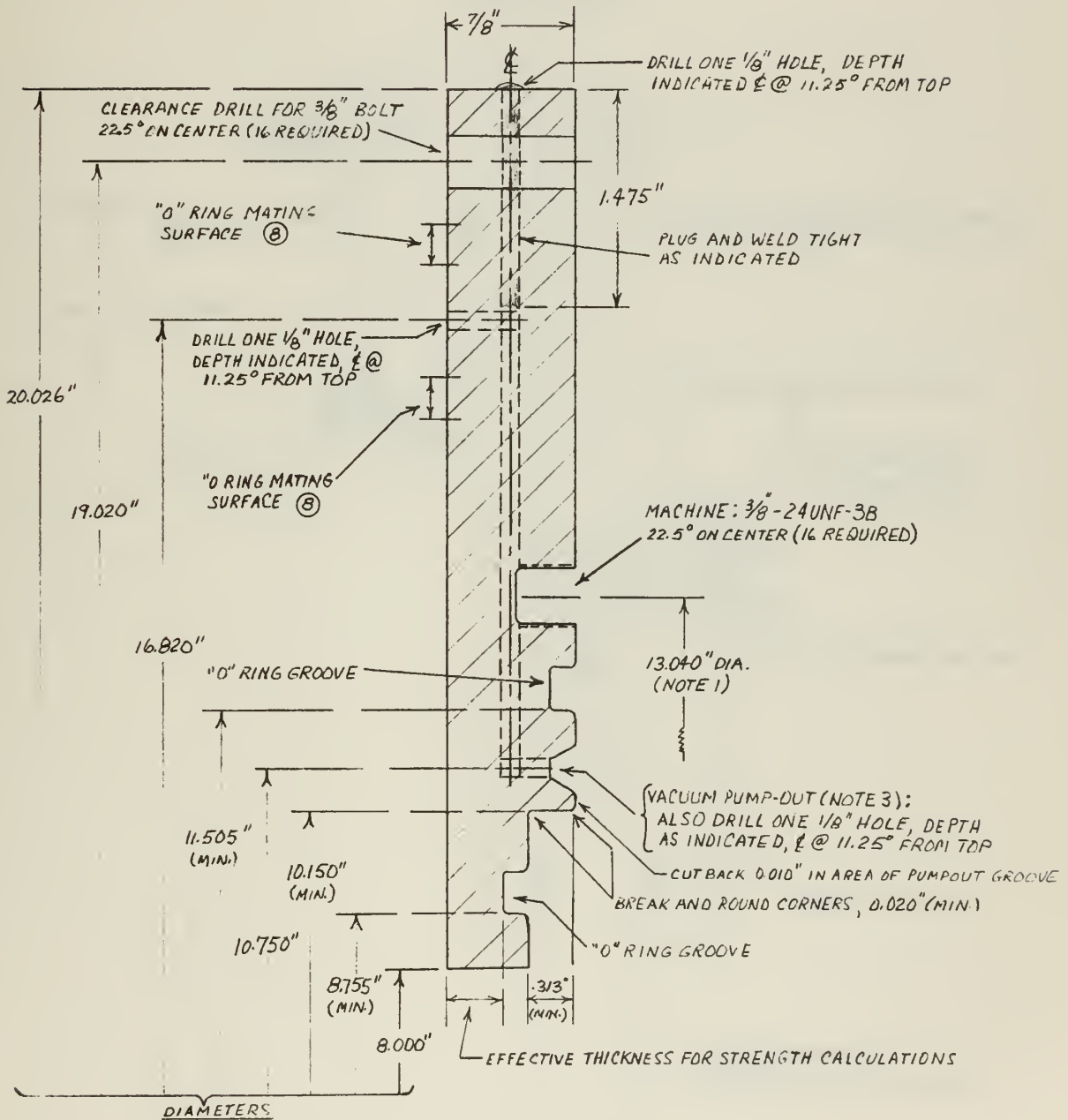
P = 1288 lbs.,

A/B = 2.15,

k = 1.54,

t = 0.392.

$$\sigma_{max} = 12,900 \text{ psi.}$$

The design factor based on the min. tensile strength is, N = 5.43, and
the design is conservative.

FIGURE 10
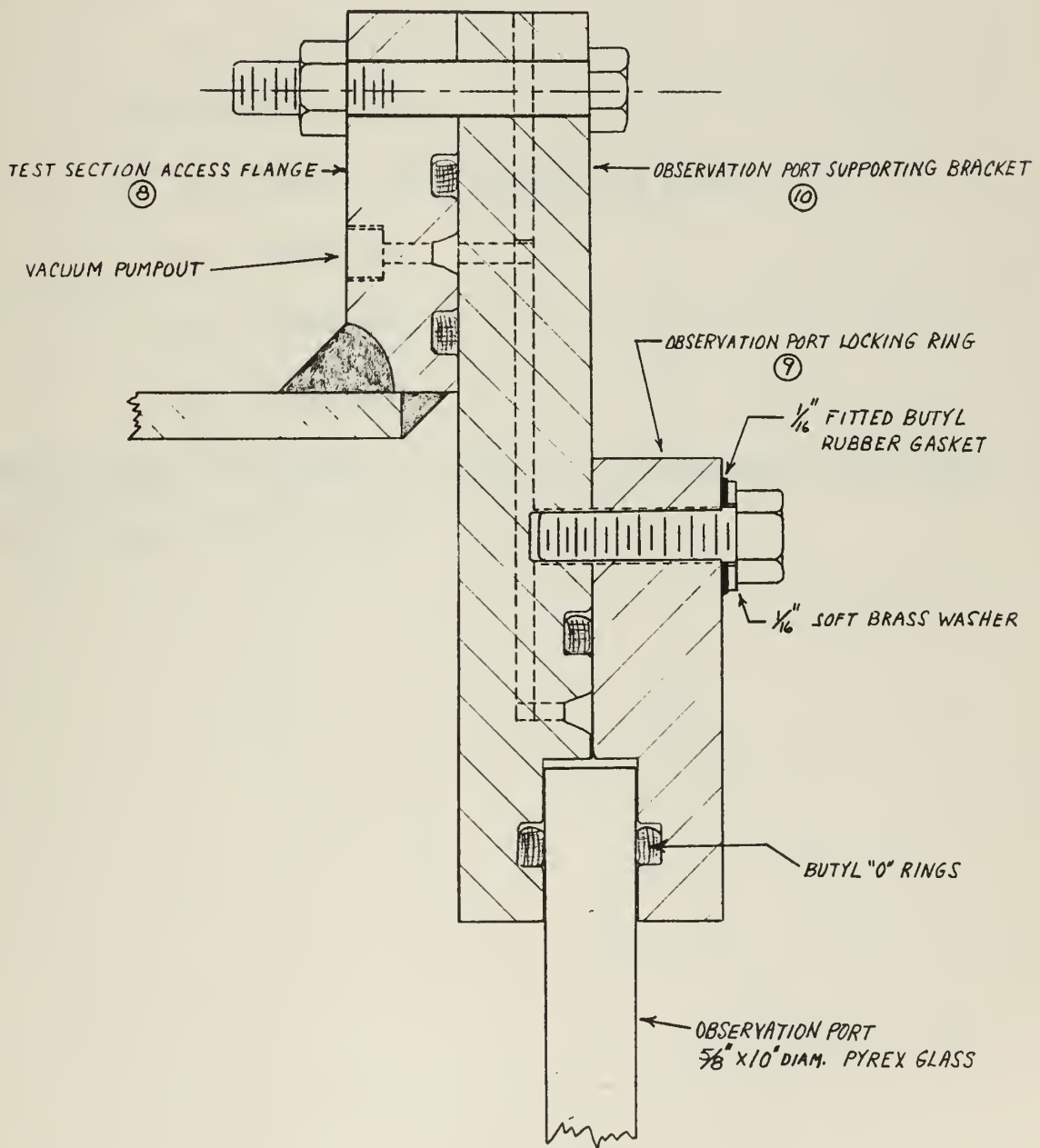
OBSERVATION PORT SUPPORTING BRACKET

NOTES:
1. Machined 3/8" - 24UNF - 3A bolts to fit mating hole in observation port locking ring. See Figs. 9 and 11. Assemble.
2. "O" Ring Groove dimensions shown in Fig. III
3. Vacuum Pump-Out Groove dimensions given in Fig. 5
4. Tolerance on all min. dimensions is + 0.005 in.
5. Tolerances not shown are ± 0.005 in.

FIGURE 11

TEST SECTION ACCESS-OBSERVATION PORT ASSEMBLY

TEST SECTION ACCESS FLANGE
⑧

OBSERVATION PORT SUPPORTING BRACKET
⑩

VACUUM PUMPOUT

OBSERVATION PORT LOCKING RING
⑨

$\frac{1}{16}$" FITTED BUTYL RUBBER GASKET

$\frac{1}{16}$" SOFT BRASS WASHER

BUTYL "O" RINGS

OBSERVATION PORT
$\frac{5}{8}$" X 10" DIAM. PYREX GLASS

E. Cooling and Vacuum Connection Flanges

This flange is shown in half-section in Fig. 12. Using the data obtained from Table II and following the procedure of Part A of this section, the values below were obtained:

(a) For operating conditions:

$$M_o = 92.4 \text{ in.-lbs.}$$

(b) For gasket seating:

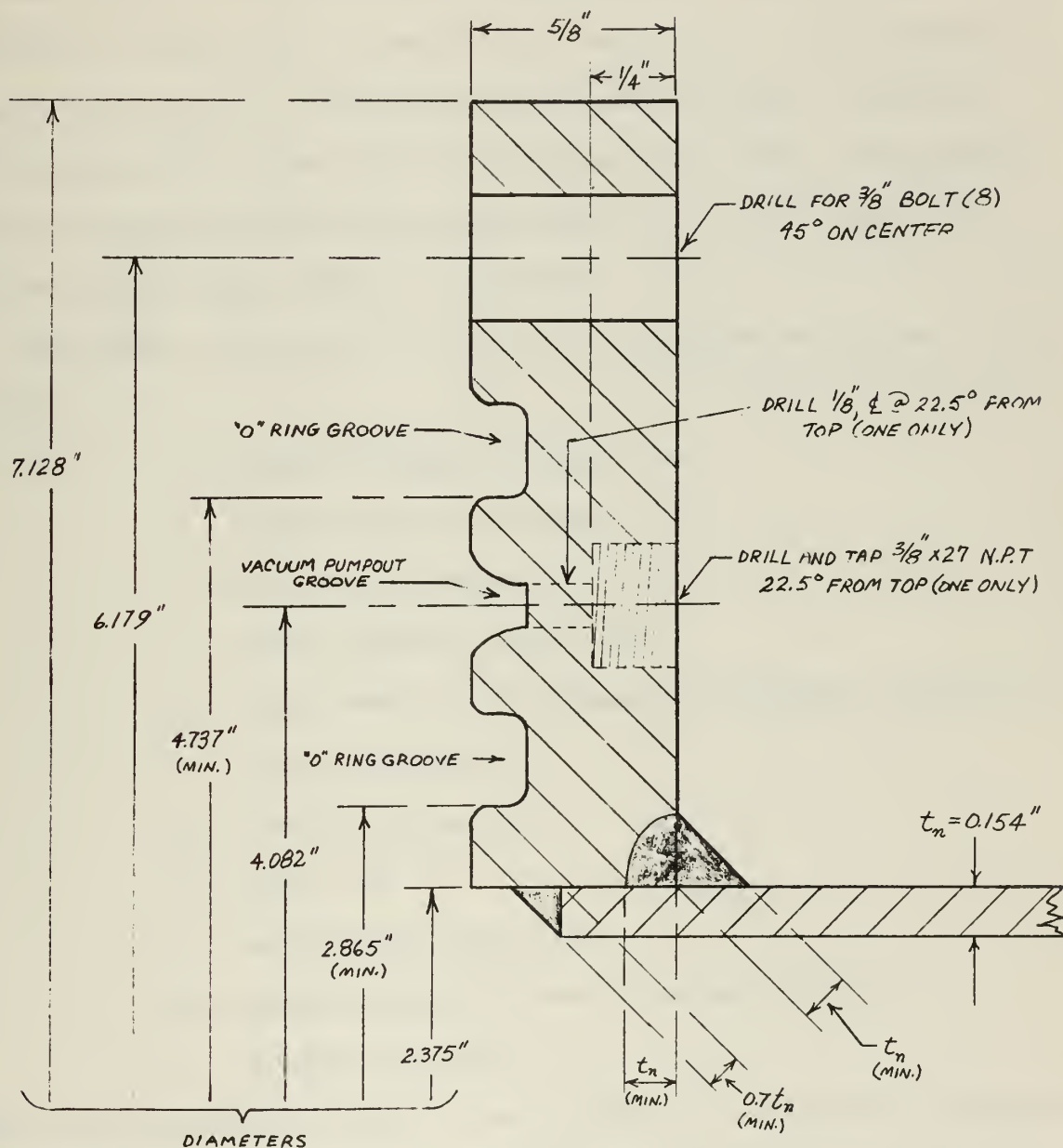$$M_o = 5,300 \text{ in.-lbs. (controlling design)},$$

$$S_H = 21,000 \text{ psi},$$

$$S_R = 9,670 \text{ psi},$$

$$S_T = 8,980 \text{ psi.}$$

Shearing stress in the weld, $S_s = 128$ psi, is conservative. The design of this flange is substantiated.

FIGURE 12
COOLING and VACUUM CONNECTION FLANGES

DRILL FOR ⅜" BOLT (8)
45° ON CENTER

DRILL ⅛", ₵ @ 22.5° FROM
TOP (ONE ONLY)

DRILL AND TAP ⅜" X 27 N.P.T
22.5° FROM TOP (ONE ONLY)

"O" RING GROOVE

VACUUM PUMPOUT
GROOVE

"O" RING GROOVE

$t_n = 0.154"$

7.128"

6.179"

4.737"
(MIN.)

4.082"

2.865"
(MIN.)

2.375"

t_n
(MIN.)

$0.7 t_n$
(MIN.)

t_n
(MIN.)

DIAMETERS
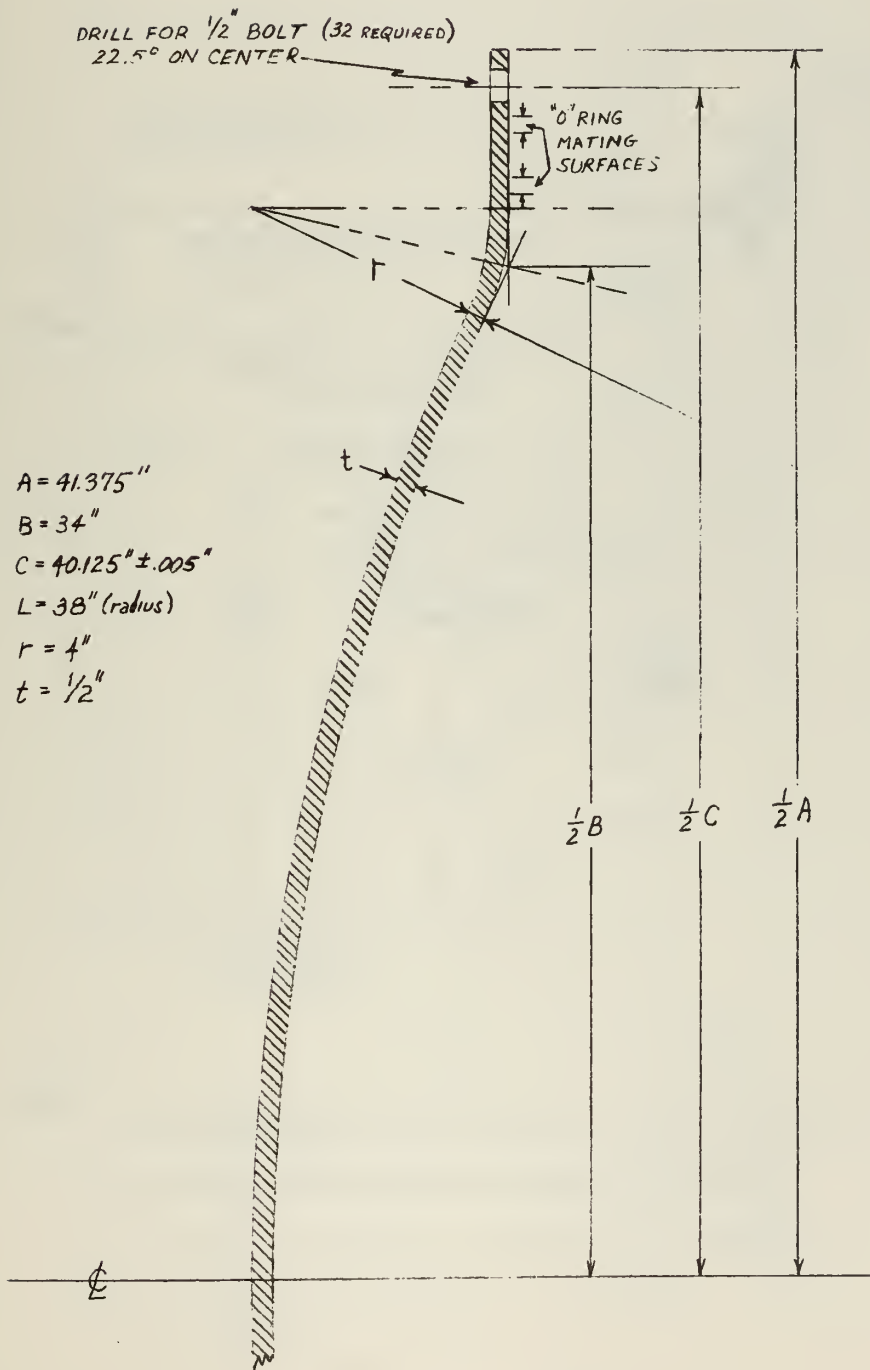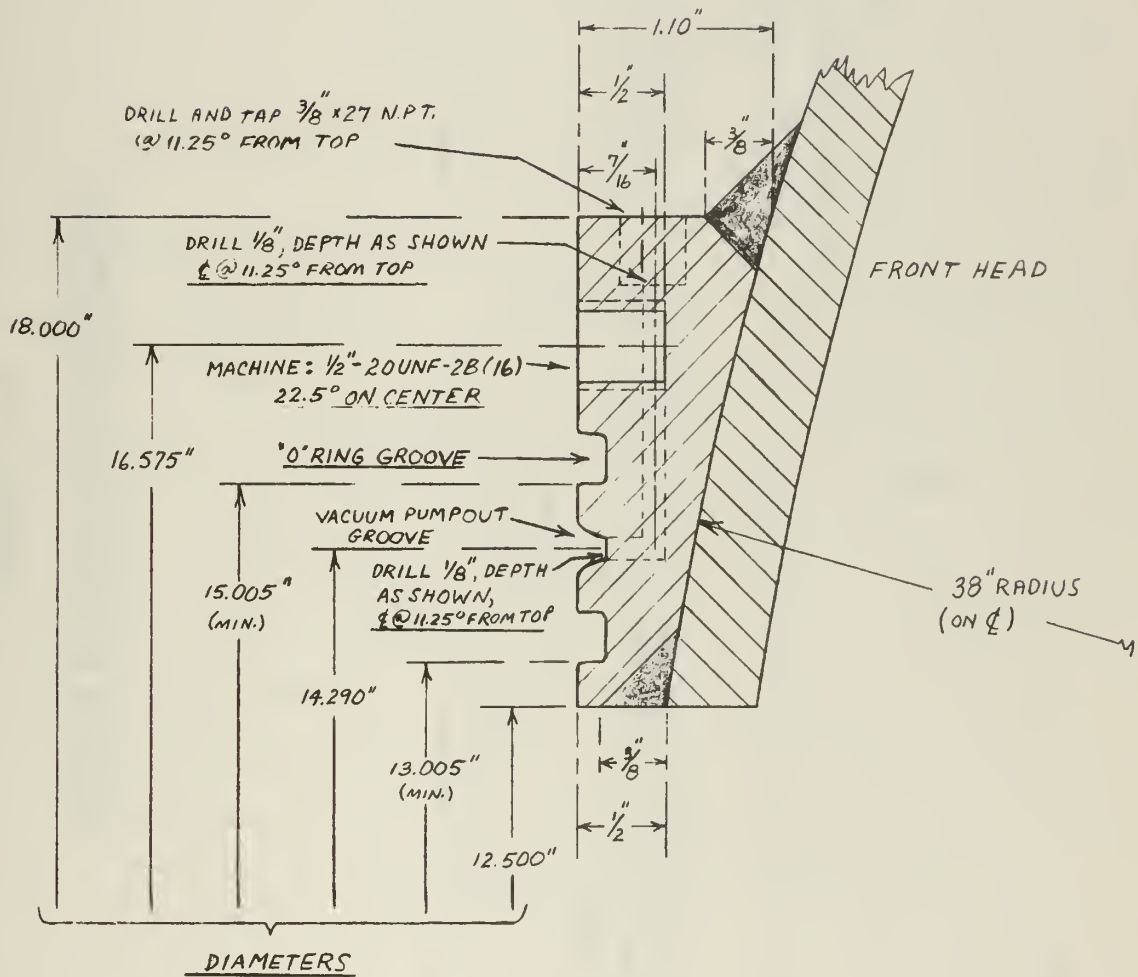
NOTES:
1. Vacuum Pump-Out Groove dimensions given in Figure 5
2. "O" Ring Groove dimensions shown in Table III
3. Tolerance on all min. dimensions is + 0.005 in.
4. Tolerances not shown are ± 0.005 in.
5. See Figure 6 for end view

6. Design of the Heads

A sketch of the basic head design is shown in Fig. 13. The opening for nozzle access distinguishes the forward (or front) head from that of the after (or rear) head. A sketch of the nozzle access ring is shown in Fig. 14, and Fig. 15 is the nozzle access assembly drawing. Both heads will be spherically dished Type 347 stainless steel covers, with integral flanges machined to act as the contact surface for the "O" rings of the section joining flanges (Section 5, Part A).

The symbols defined below are used in the formulae and figures of this section:

A = outside diameter of flange, in.

B = inside diameter of flange, in.

C = bolt circle diameter, in.

L = inside spherical radius, in.

M_o = total moment on the flange as determined in Section 5, Part A, in.-lbs.

P_e = design pressure, psi

Q = factor, in.

r = outside dish radius, in.

S = maximum allowable stress value for plate, psi

t = thickness of plate, in.

Using the values shown on Fig. 13, and following the procedure of paragraph UA-6, Appendix I to Section 8 of the Code, the calculations below were made to find the minimum required thickness:

From UA-6, (b), (3):

(a) $t = (5 P_e L)/6S = 0.0253$ in.

(b) $t = Q \left[1 + \left(1 + \dfrac{7.5\,M_o}{PQBL} \right)^{1/2} \right]$;

$$Q = \frac{P_e\,L}{4\,S} \left[\frac{1}{1 + \dfrac{6(C - B)}{(C + B)}} \right],$$

$Q = 0.00506$ in.

From Part A of Section 5, $M_o = 42{,}062$ in.-lbs., therefore,

$t = 0.292$ in.

(c) The larger value of (a) or (b) is the minimum required thickness of head.

Use 1/2 inch head thickness to provide additional metal for reinforcement of openings, and stiffening against deflections.

FIGURE 13

BASIC HEAD DESIGN

DRILL FOR ½" BOLT (32 REQUIRED)
22.5° ON CENTER

"O" RING
MATING
SURFACES

r

t

A = 41.375"
B = 34"
C = 40.125" ± .005"
L = 38" (radius)
r = 4"
t = ½"

L

$\frac{1}{2}B$ $\frac{1}{2}C$ $\frac{1}{2}A$
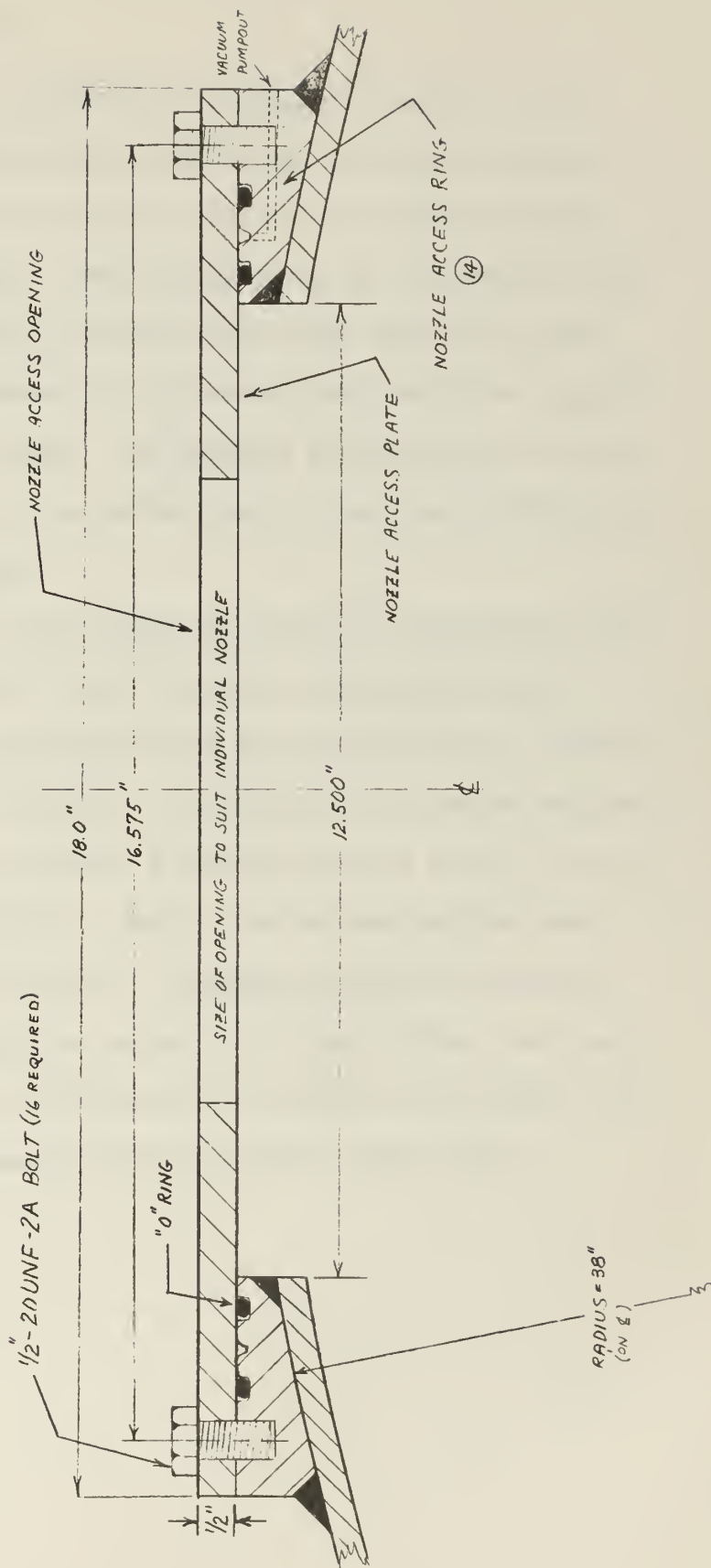
$\mathord{\mathbb{C}}$

FIGURE 14

NOZZLE ACCESS RING

NOTES:
1. Vacuum Pumpout Groove dimensions given in Figure 5
2. "0" Ring Groove dimensions shown in Table III
3. Tolerance on all min. dimensions is + 0.005 in.
4. Tolerances not shown are ± 0.005 in.
5. See Figure 6 for end view

FIGURE 15

NOZZLE ACCESS ASSEMBLY

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7. Vessel Supporting Structure

In order to design the supporting structure, the weights of the system components were calculated from known data or assumed configurations. The heaviest section of the structure is the cryopump section, its total weight being 1000 pounds. The dimensions of the supports are such that either one safely carries the total load, there being two cradle supports per section. Consequently, the assembly has sufficient rigidity and capacity for extraneous loads. The supports will rest on an I-beam track, enabling each section to be pulled clear of the rest of the system upon unbolting of the flanges.

Figure 16 is a sketch of the supporting structure showing all dimensions and components. Standard sized components were chosen, and it is intended that plain carbon structural steel be used. The stress analysis included column effect where apropos. Calculations for combined stresses indicated that the vertical legs have a design factor, $N = 3.67$, by the "maximum shear stress" theory /7/. The tie rod between the legs had a maximum equivalent stress of 874 psi. Although the apparent factor of safety here seems rather high, the intent is to limit deflections toward the bottom of the structure, as any bowing of the legs will tend to bind the "free running" of the support members on their I-beam track.

FIGURE 16

VESSEL SUPPORTING STRUCTURE

18"

30°

60°

4 U 7.25

16.4"

2" STD. (SCHED 40) PIPE

24"

10"

4 U 7.25

5 I 10

36"
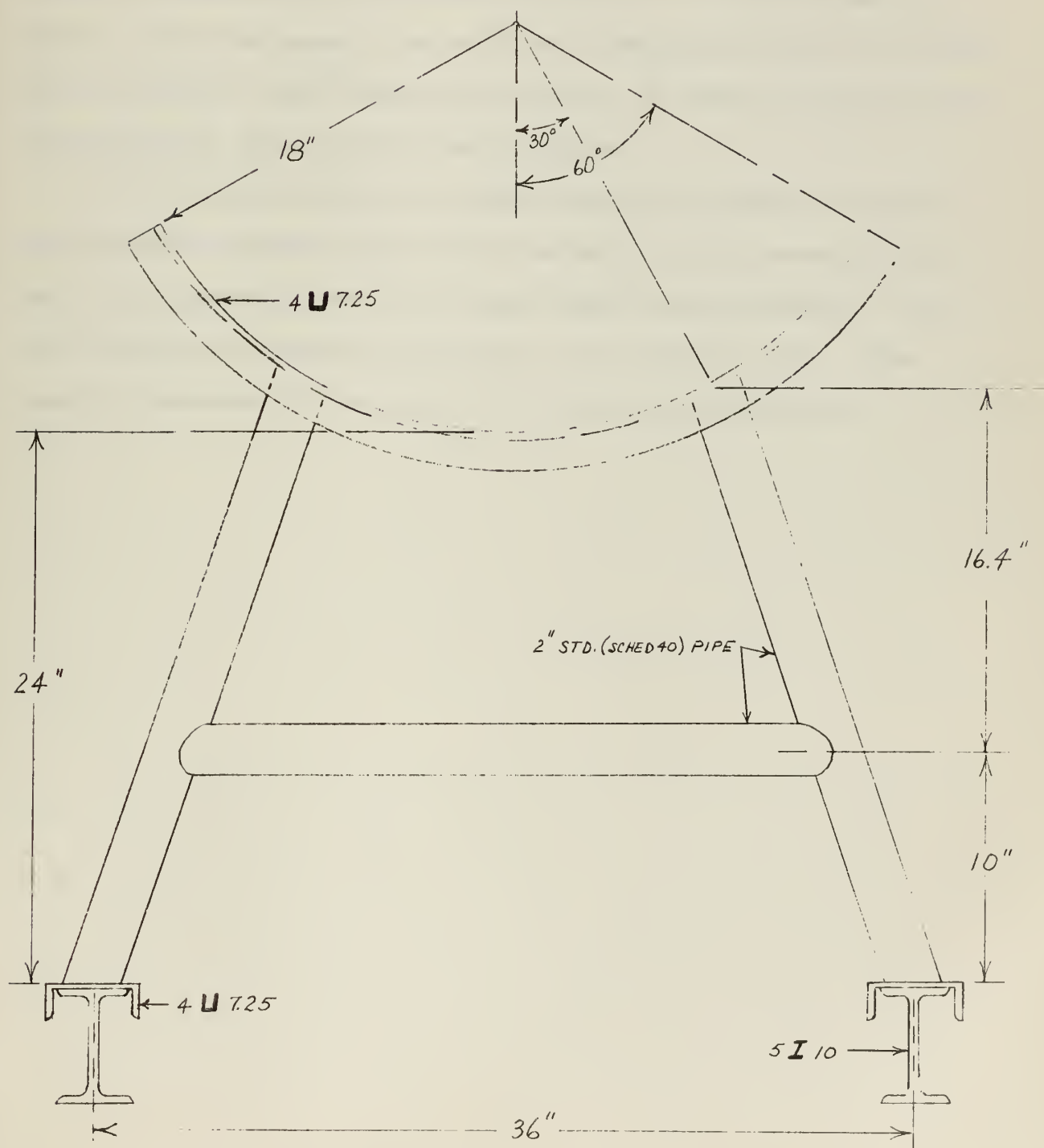
8. General Concept of Interior Component Support

The areas immediately adjacent to the end flanges and stiffening
ring in each section are much stiffer than the shell plate, and provide
convenient supporting points. The use of these areas for support of
interior components will preclude the possibility of overstressing the
shell. The bottom supports for the interior components should be centered
over the exterior vessel supporting structure. No manner of support should
be attempted in the vicinity of shell openings.

It is recommended that the internal supports be drilled and tapped
stainless steel blocks, shaped to fit the tank wall, and secured to the
wall in the desired location with a good grade of epoxy adhesive. This
will preclude the necessity for welding in the finished product. The
interior components will be bolted to the tapped support blocks.

9. Control Valve Design

By use of a throttling device, one may control the pressure in the test section of the facility. As ice forms on the cryopump surfaces, vapor pressure in the cryopump section increases. By opening the control valve, one hopes to maintain a constant pressure in the test section while the pressure in the cryopump section is increasing. The control valve will be contained in a separate housing of rectangular cross-section connected to the intercooler and cryopump sections by sleeved flanges, which mate with the tank section joining flanges as designed in Part A of Section 5. The control valve housing will be fabricated from 1/4-inch sheet and will be stiffened by a three-inch standard, 7.5-pounds-per-foot I-beam. Figure 17 is a plan view of the housing.

The housing will contain the control valve plates and their driving mechanism. The drive motor will be a 1/50 to 1/20 horsepower servo-motor whose output shaft will be coupled to a one-inch, five-threads-per-inch, Acme threaded power screw. The motor, and the bearing supports for the power screw, will be attached to the I-beam stiffener of the housing. Figures 18 and 19 show the interior configuration and give some of the important dimensions.

The valve plates will be 38-inch squares of 3/32-inch Type 347 stainless steel plate with a 36-inch diameter semi-circle cut in one edge. Each plate will be supported through its center-of-gravity by a bracket and drive nut. The drive nut will be a two-inch cube of Type 347 stainless steel, bored to take a cementable Teflon insert which will be machined to a one-inch, five-threads-per-inch, Acme threaded nut. Figure 20 shows the valve drive mechanism details. In order to keep the bottom edges of the plates

45

from swinging together, a roller assembly is installed at the center line of each plate (Figs. 18, 21).

A manual input to a servo-motor located outside the housing will provide the electrical control of the drive motor. An electronic or manual counter will count the number of revolutions of the power screw so that one may reset the control valve to such a position as to duplicate interesting phenomena occuring at some given setting of the valve. This method of drive was chosen so that there would be no moving part piercing the vacuum seal.
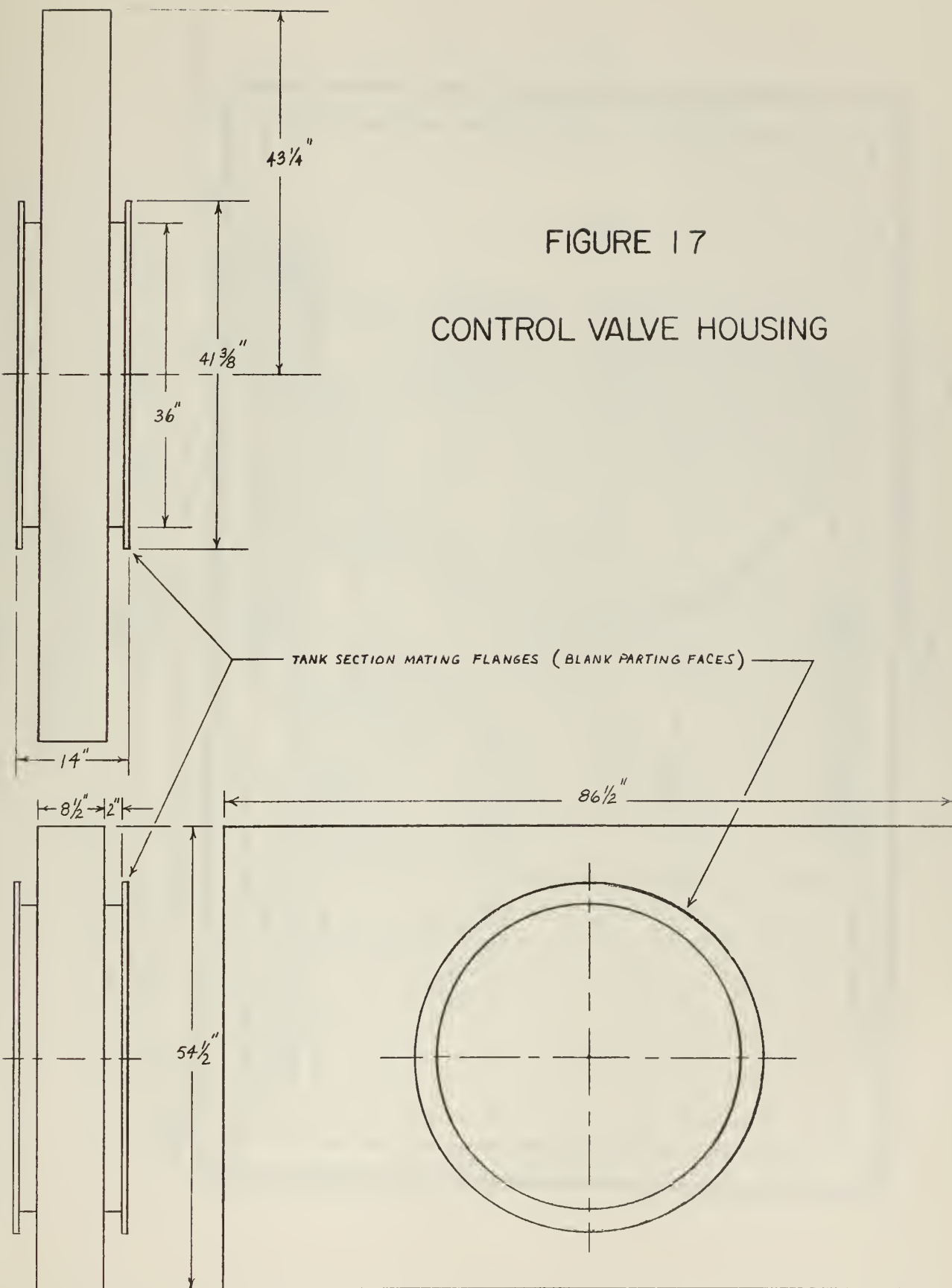
FIGURE 17

CONTROL VALVE HOUSING

TANK SECTION MATING FLANGES (BLANK PARTING FACES)

43¼"

41⅜"

36"

14"

8½" 2"

54½"

86½"

FIGURE 18

CONTROL VALVE

48
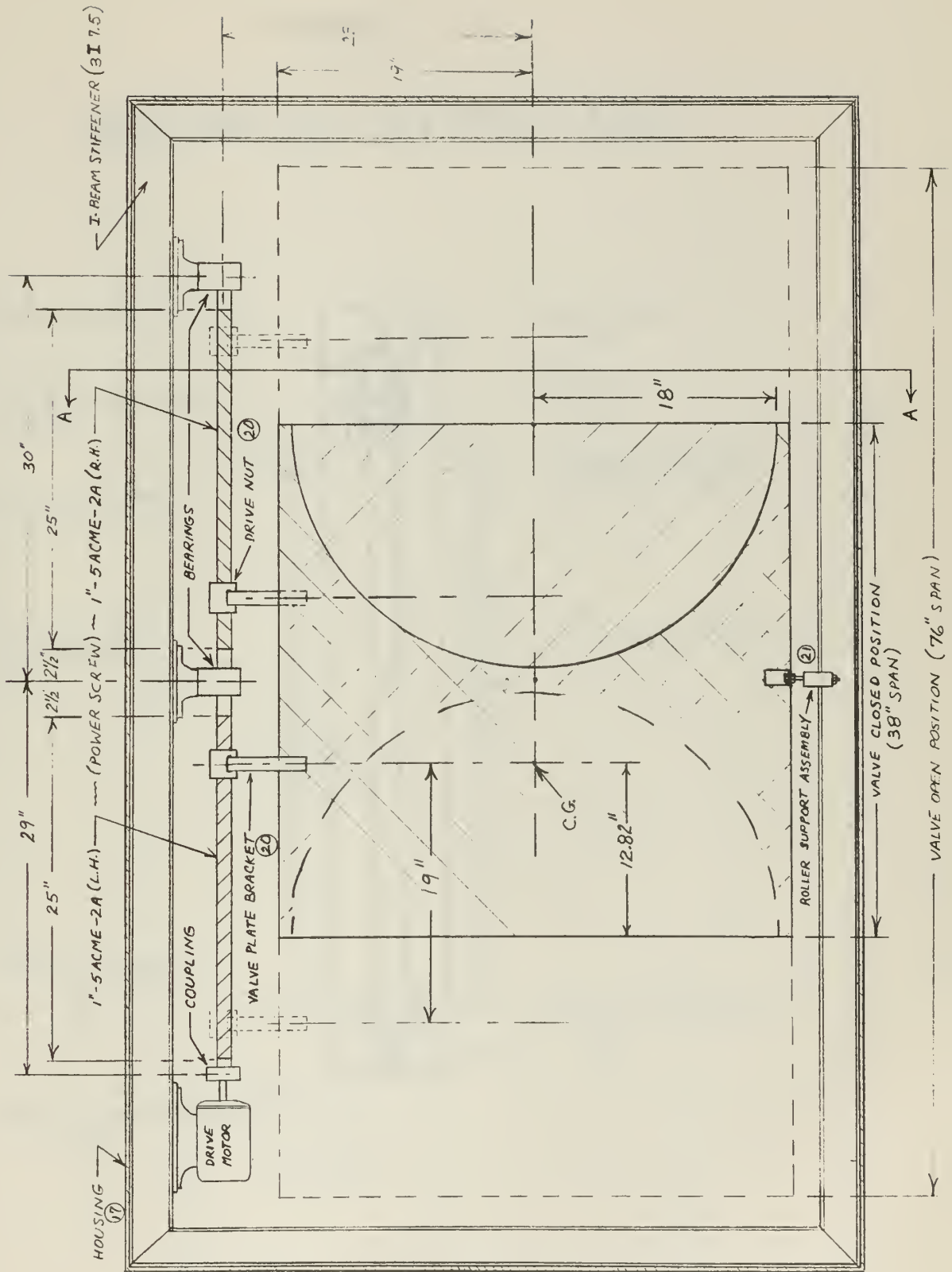
FIGURE 19

SECTION A-A, CONTROL VALVE

BEARING HOUSING

DRIVE MOTOR
(HIDDEN)

VALVE PLATE BRACKET
(20)

POWER SCREW
(20)

DRIVE NUT
(20)

CRYOPUMP SECTION
MATING FLANGE

INTERCOOLER SECTION
MATING FLANGE

VALVE PLATES ($\frac{3}{32}$" PLATE)

ROLLER SUPPORT
(21)

HOUSING ($\frac{1}{4}$" PLATE)
(17)

I-BEAM STIFFENER (3 I 7.5)
(21)
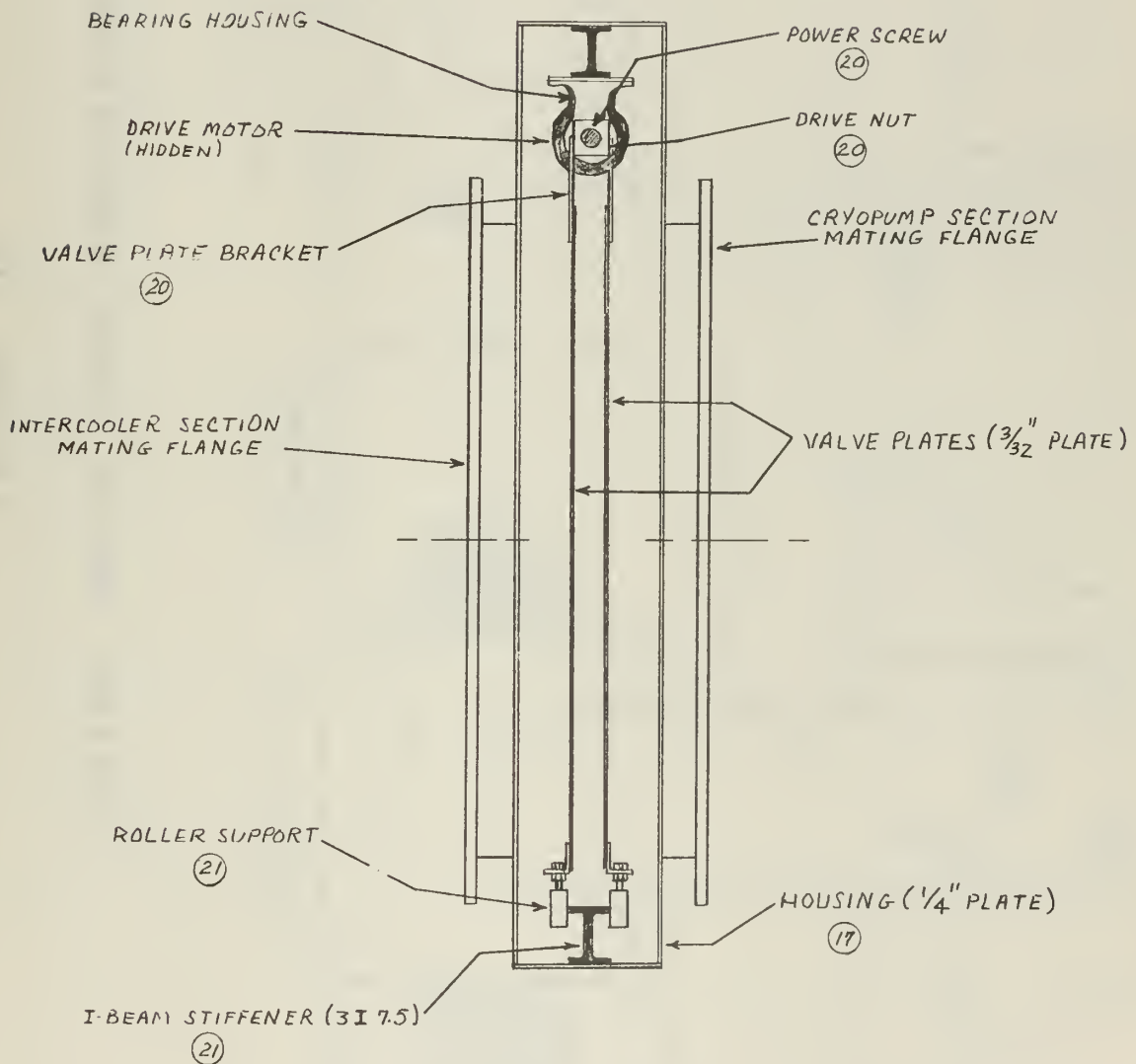
FIGURE 20

CONTROL VALVE DRIVE MECHANISM

50
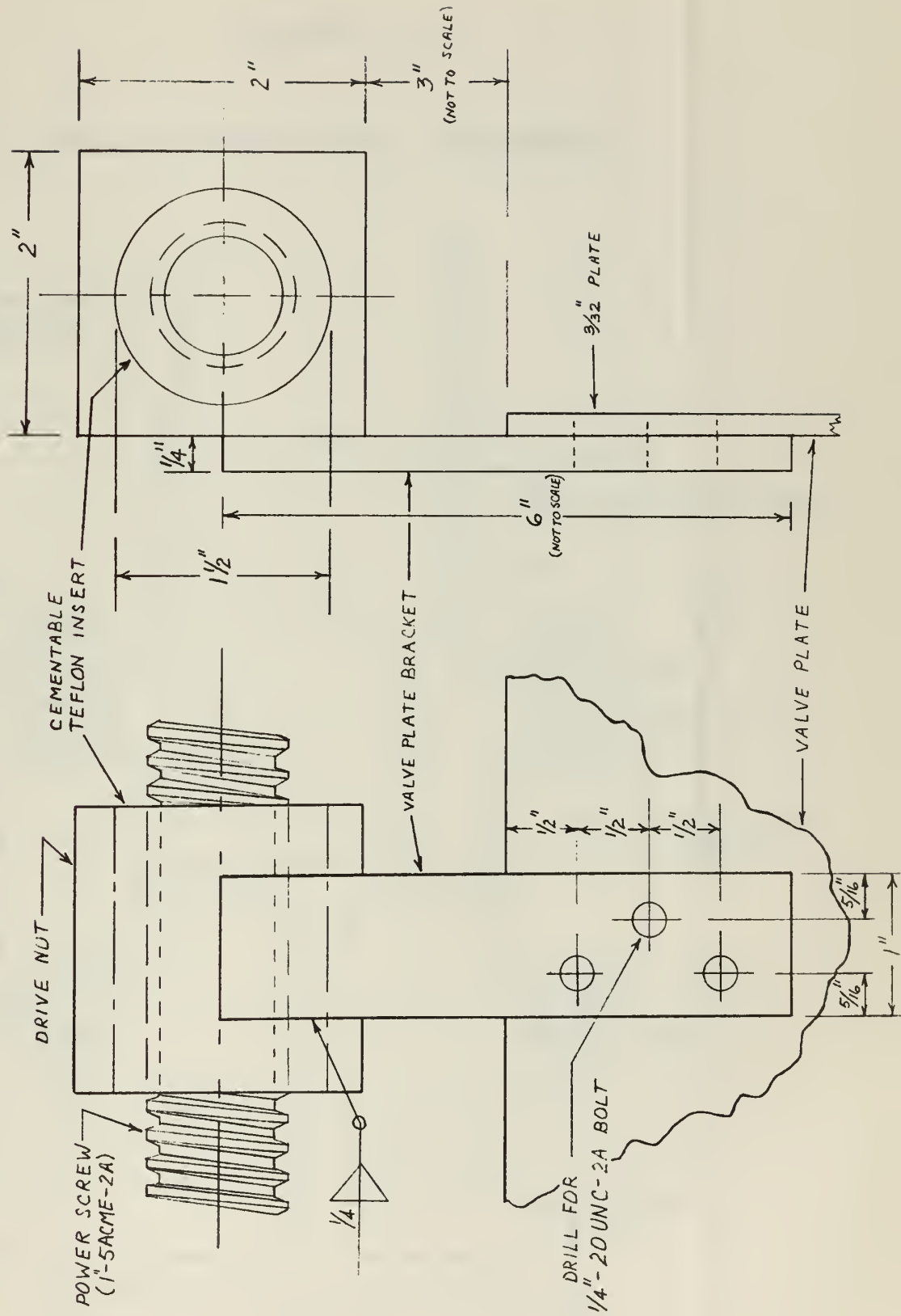
FIGURE 21

ROLLER SUPPORT ASSEMBLY

VALVE PLATE
(³⁄₃₂" PLATE)

1 ¹³⁄₁₆"

L 2 × 1½ × ⅛
(BOLT OR WELD TO VALVE PLATES)

⅜-16 UNC-2A BOLT

2"

¾"

LOCK WASHER

½"

1¼"

⅜-16 UNC-2B NUT

TEFLON ROLLER
O.D. = 1"

1½"

⅜"-16 UNC-2B NUT

WASHER

COTTER PIN

3½"

¼

HOUSING STIFFENER
3 I 7.5
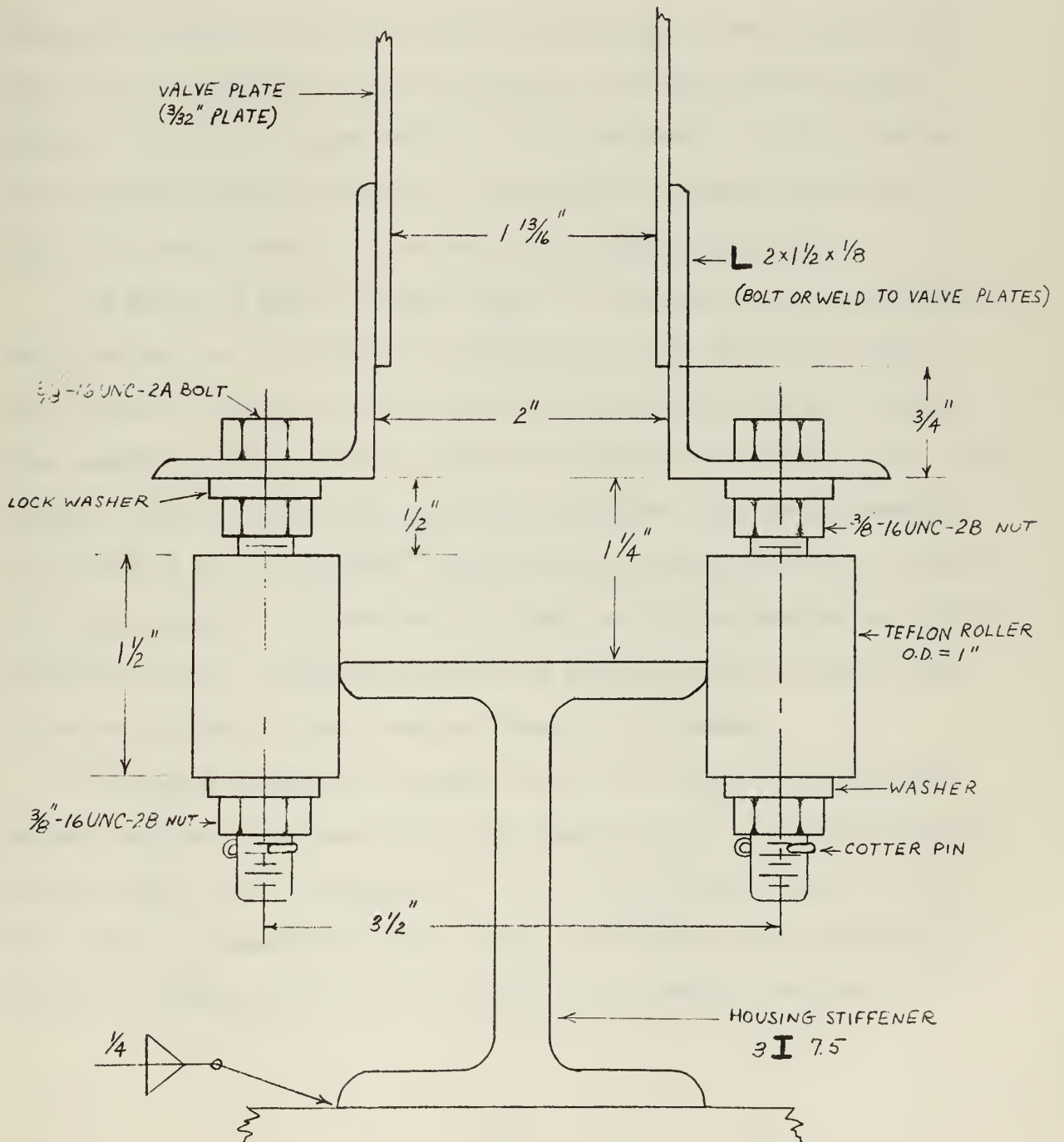
10. Shut-Off Valve Design

In order to obtain the maximum utility from the proposed facility, it becomes necessary to allow for change of the model under test or change of the instrumentation in the test section without charging the entire system with raw air. To satisfy this condition, a shut-off valve has been designed to isolate the test section from the rest of the facility. The valve body is a separate housing connected to the test and intercooler sections by sleeved flanges which mate with the section joining flanges as designed in Part A of Section 5. The valve is a gate type valve. Figures 22 and 23 show the valve body (or housing) construction.

The gate is a built up wedge (Figs. 25, 26) which is hung by brackets and drive nuts on a two-inch, four-threads-per-inch, Acme power screw. The drive nuts are four inch cubes of Type 347 stainless steel, bored to take cementable Teflon inserts, which are machined to match the power screw threads. Figure 28 shows the gate drive mechanism. The power screw will be coupled to a 1/2 horsepower servo-drive motor which receives its electrical drive signal from a servo-motor located outside the housing and operated by manual input. The drive motor and the bearing supports for the power screw are attached to the I-beam stiffener of the housing.

The seating surface for the gate is a 1/2-inch thick ring of butyl rubber, secured to the rear wall of the housing with a good grade of epoxy adhesive (Fig. 24). The ring has an I.D. of 36.5 inches and an O.D. of 38.0 inches. A cementable Teflon insert, secured with epoxy adhesive in a channel iron (Fig. 27), will act as the bottom guide for the gate.

FIGURE 22

SHUT-OFF VALVE
HOUSING
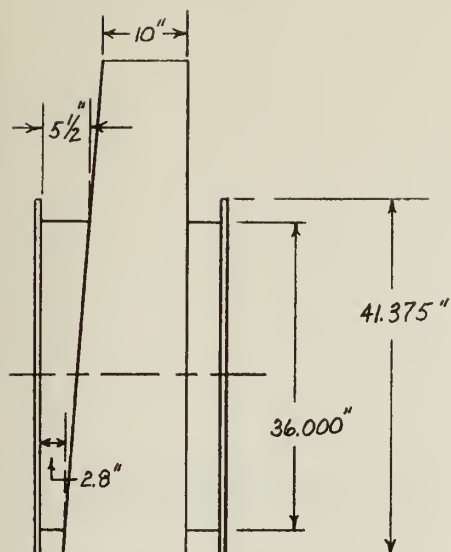
NOTE: BOTH FLANGES MATE WITH SECTION JOINING FLANGES ⑦
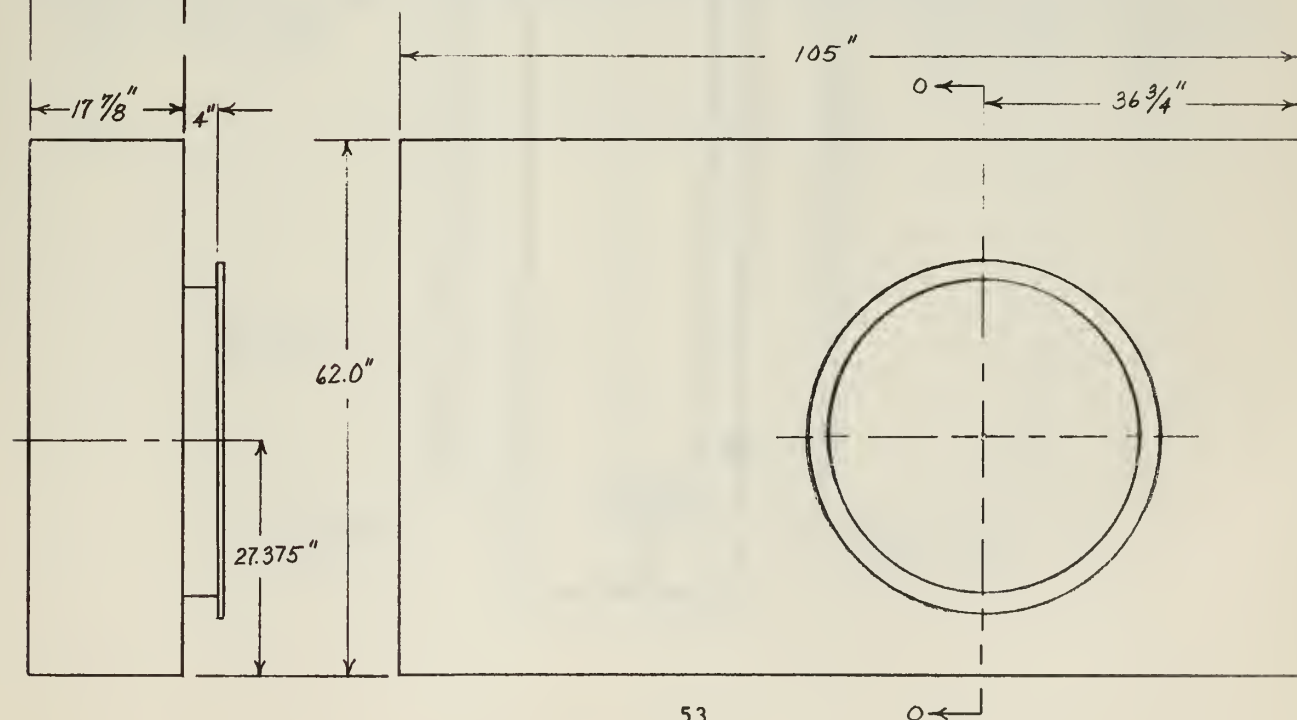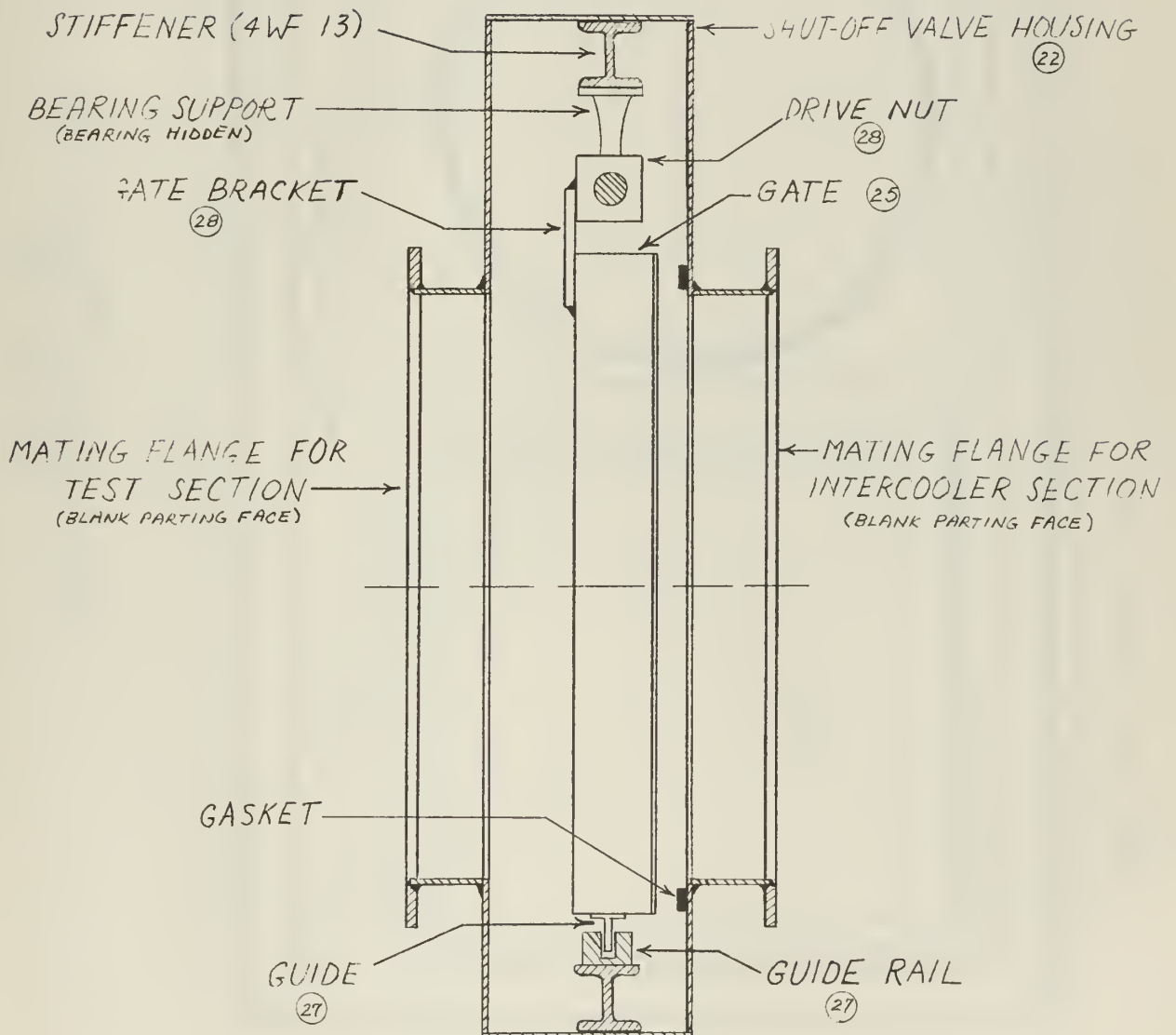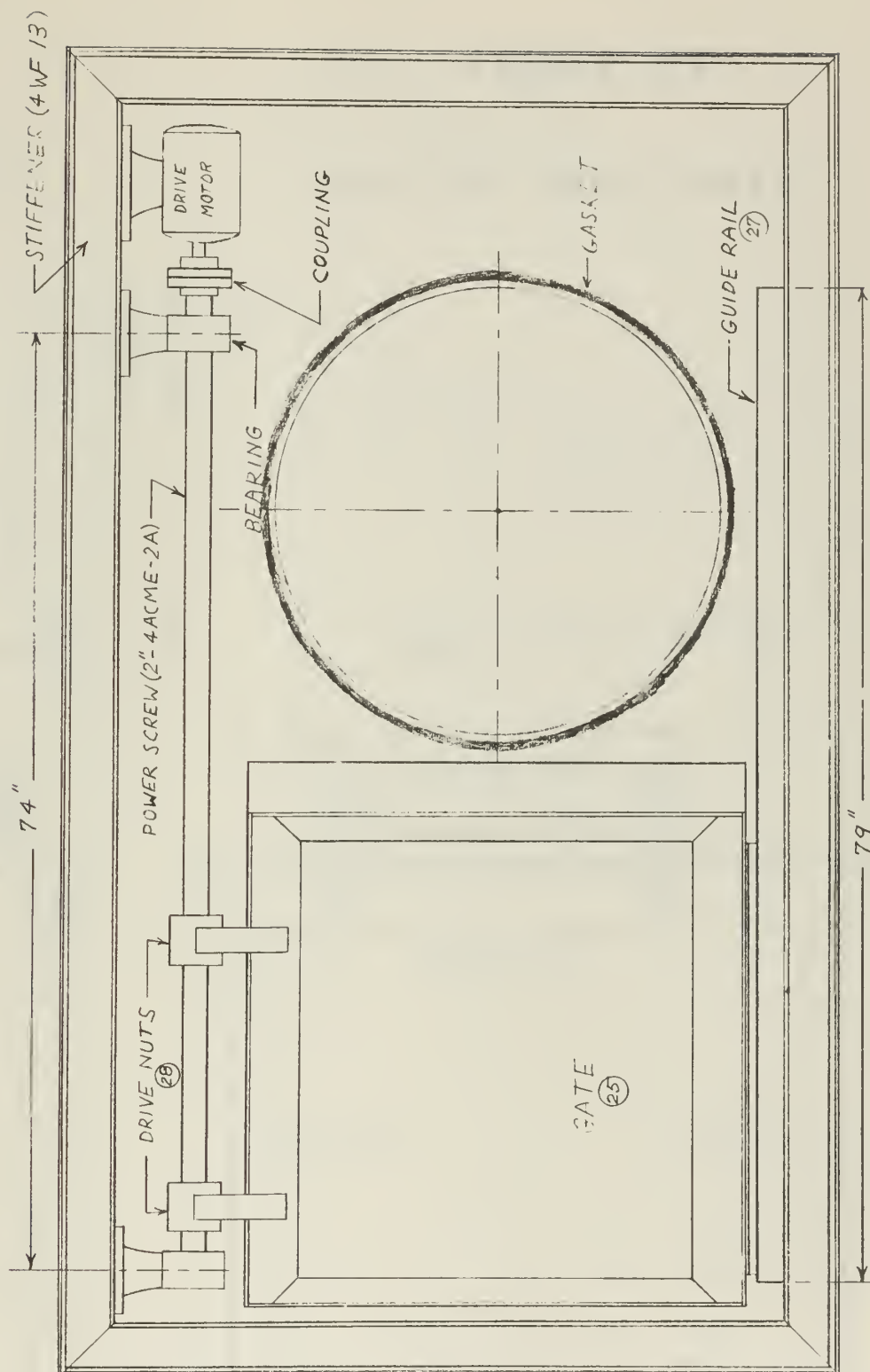WITH BLANK PARTING FACE
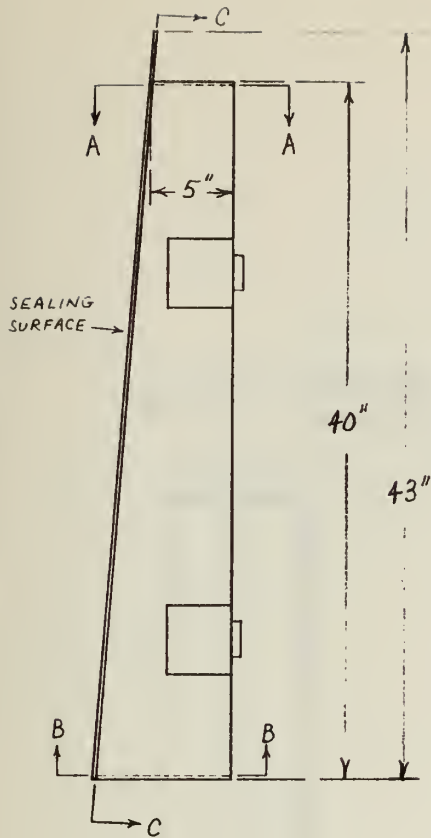
53

FIGURE 23

SHUT-OFF VALVE
SECTION O-O

STIFFENER (4WF 13)

BEARING SUPPORT
(BEARING HIDDEN)

GATE BRACKET
(28)

SHUT-OFF VALVE HOUSING
(22)

DRIVE NUT
(28)

GATE (25)

MATING FLANGE FOR
TEST SECTION
(BLANK PARTING FACE)

MATING FLANGE FOR
INTERCOOLER SECTION
(BLANK PARTING FACE)

GASKET

GUIDE
(27)

GUIDE RAIL
(27)

FIGURE 24
SHUT-OFF VALVE PLAN

55

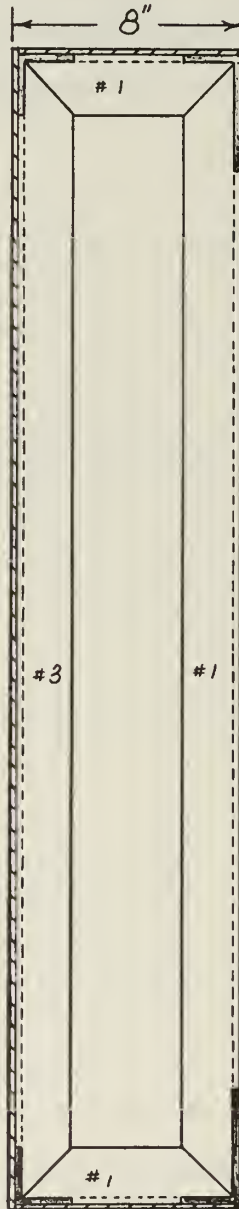FIGURE 25

SHUT-OFF VALVE GATE

NOTE: ALL PLATE IS ¼" THICK
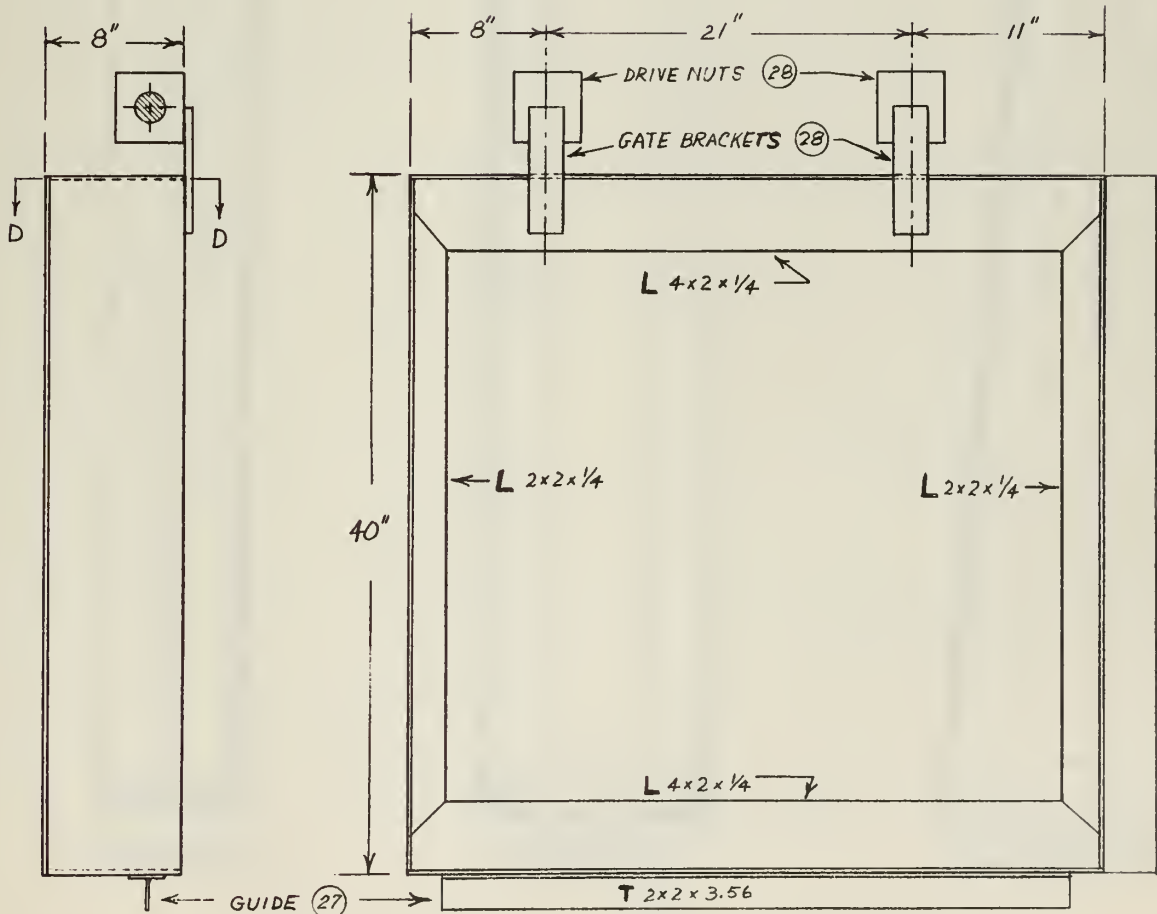
FIGURE 26a
GATE SECTIONS

#1 = 2×2×¼ ANGLE IRON : ANGLE = 90°
#2 = 2×2×¼ ANGLE IRON : ANGLE = 94.3° } 26b
#3 = 2×2×¼ ANGLE IRON : ANGLE = 85.7°

5"

8"

SEATING PLATE →
(43"×40"×¼")

#1   #2

#3   #1

#1   #1

#1

#1

#1

SECTION A-A

SECTION B-B
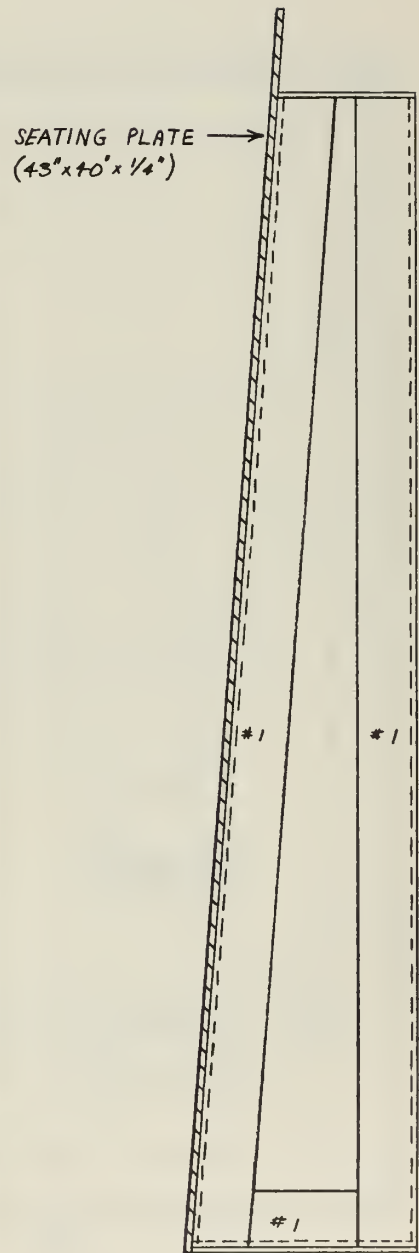
SECTION D-D

FIGURE 26 b
GATE SECTION C-C
SEALING SURFACE
(SEATING PLATE REMOVED)

TOP PLATE (1/4")

L 2×2×1/4
(∢ = 94.3°)

X
X

94.3°±0.3°

SECTION X-X

85.7°±0.3°

SECTION Y-Y

Y
Y

(∢ = 85.7°)

L 2×2×1/4

BOTTOM PLATE (1/4")
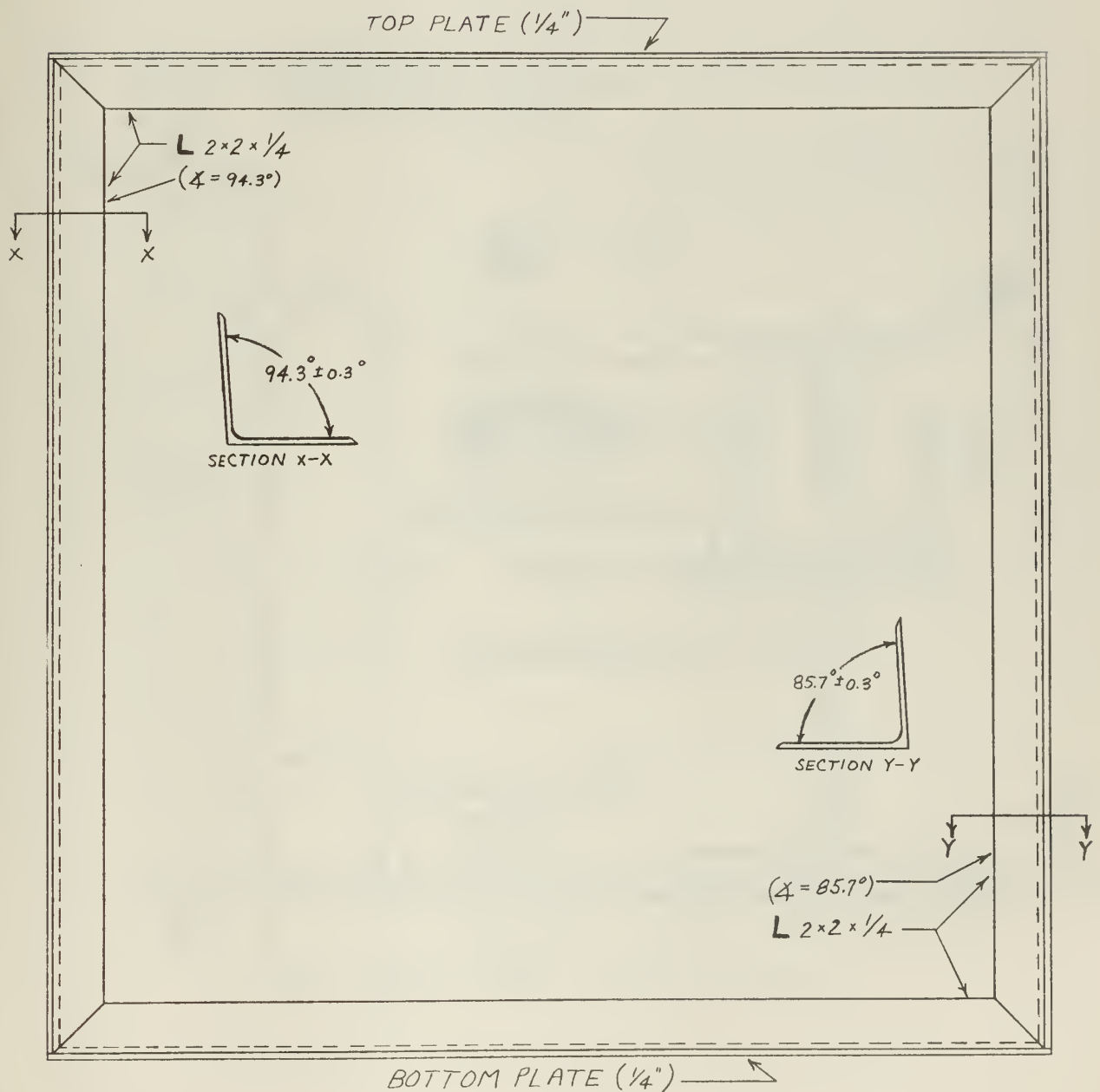
FIGURE 27

VALVE GUIDE ARRANGEMENT
SHUT-OFF VALVE

SLEEVE FOR TEST SECTION
MATING FLANGE ($D_o = 36"$)

¼"

¼"

GATE
㉔㉕

GUIDE ($T\ 2 \times 2 \times 3.56$)

CEMENTABLE
TEFLON BEARING
(GROOVE WIDTH = 9/32")
(GROOVE DEPTH = 1¼")

GUIDE RAIL ($3 \times 1\frac{7}{8}$ U9)

5/16

9.125"

7.125"

STIFFENER ($4\,WF\,13$)

VALVE HOUSING
㉒ ㉓

5/16
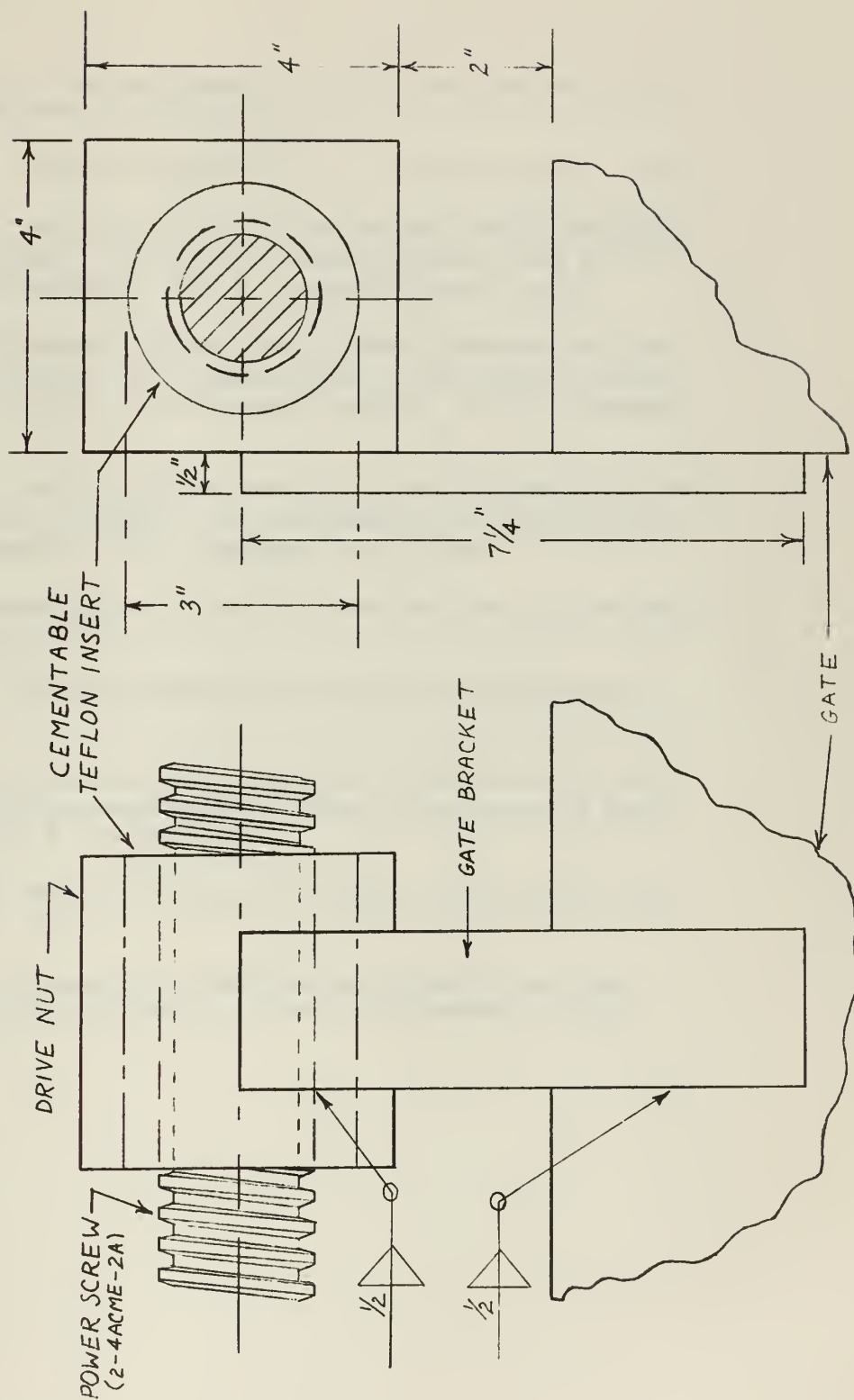
¼"

1"

1¼"

2⅛"

4"

7.375"

59
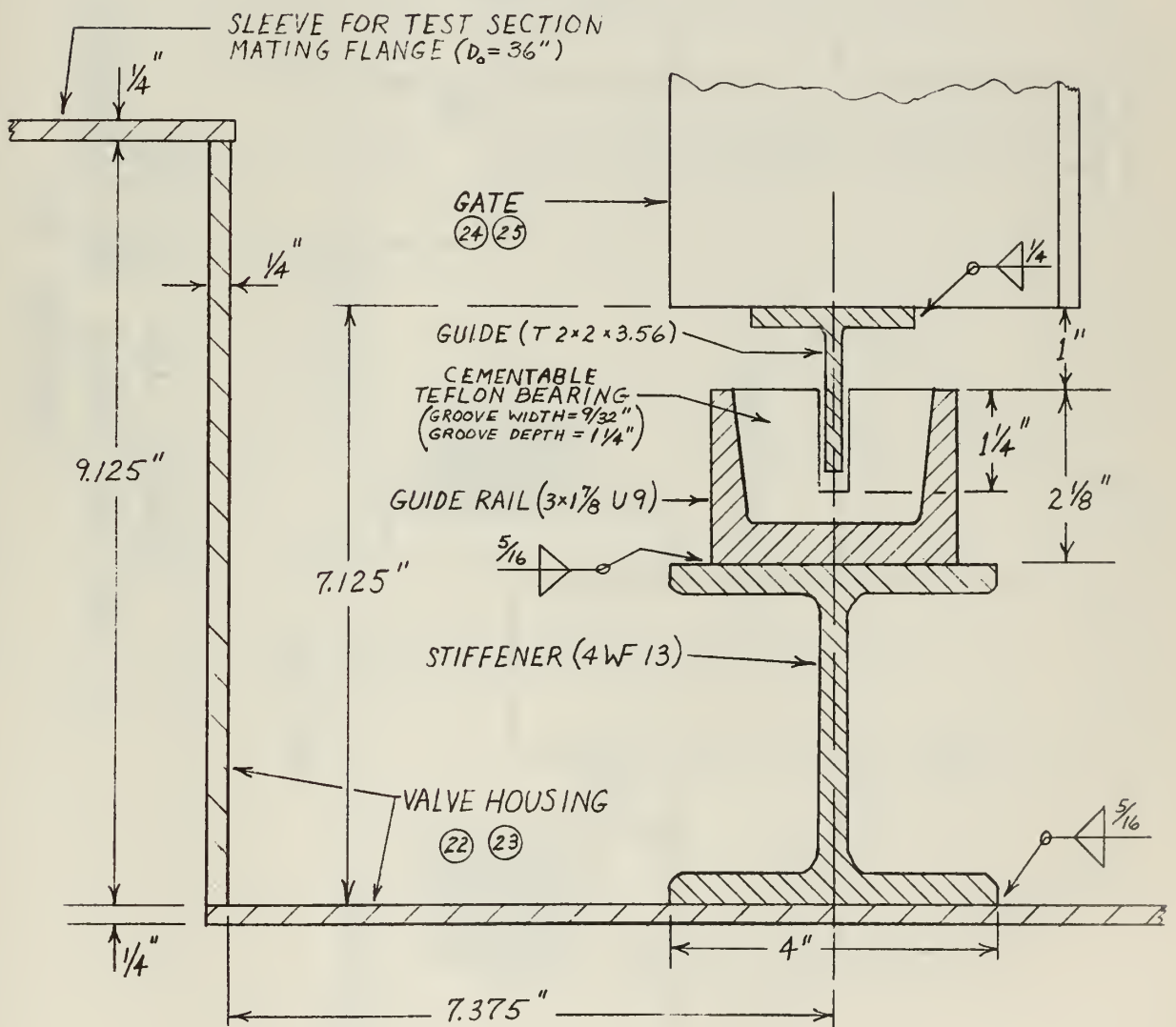
FIGURE 28

VALVE DRIVE MECHANISM
SHUT-OFF VALVE

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APPENDIX I

DESIGN OF THE TANK SECTION JOINING FLANGE

(Part A of Section 5)

Using the dimensions taken from Fig. 7, and following the procedure of Appendix II to Section 8 of the Code, the calculations below are made to substantiate the design of the section joining flange:

1. A_b = (number of bolts, N)x(stress area per bolt, A_s)

    A_s = 0.1374 sq. in. /7/

    N = 32

    A_b = 4.3968 sq. in.

2. $G_m = (G_1 + G_2)/2$

    G_1 = 36.500 + 0.210 = 36.710 in.

    G_2 = 38.563 + 0.210 = 38.773 in.

    G_m = 37.741 + tolerance

    G_m = 37.745 in.

3. W_{m2} = 3.14 G_m Q

    Q = 1.33 x 15 lbs. per lineal inch of circumference /8/

    W_{m2} = 4,740 lbs.

4. From Table UHA-23, Section 8 of the Code:

    $S_a = S_b$ = 15,000 psi

    S_f = 17,500 psi

    S_n = 16,000 psi

5. $A_{m2} = W_{m2}/S_a$

    A_{m2} = 4,740/15,000 = 0.316 sq. in.

6. t = (total flange thickness) - (depth of "O" ring cut)

    t = 0.750 - 0.170 = 0.580 in.

7. h = length of weld leg (Fig. UA-48 of Code)

    h = 0.355 in.

8. g_o = thickness of plate (Fig. UA-48 of Code)

$g_o = 0.250$ in.

9. $h_o = (B\ g_o)^{1/2} = (36.0 \times 0.25)^{1/2} = 3.00$ in.

10. g_1 = thickness of plate + length of weld leg (Fig. UA-48 of Code)

    $g_1 = 0.250 + 0.355 = 0.605$ in.

11. $K = A/B$

    $K = 41.375/36.0 = 1.15$

12. From Fig. UA-51.1 of the Code:

    $T = 1.85$

    $U = 17.0$

    $Y = 15.0$

    $Z = 8.0$

13. From Fig. UA-51.2 of the Code:

    $F = 0.902$

14. From Fig. UA-51.3 of the Code:

    $V = 0.41$

15. From Fig. UA-51.6 of the Code:

    $f = 4.35$

16. $d = \dfrac{U\ h_o\ (g_o)^2}{V}$

    $d = \dfrac{17.0 \times 3.0 \times 1/16}{0.41} = 7.78$

17. $e = F/h_o$

    $e = (0.902)/(3.0) = 0.3$

18. $L = \dfrac{(t\ e + 1)}{T} + \dfrac{(t)^3}{d}$

    $L = \dfrac{(0.58 \times 0.3 + 1)}{1.85} + \dfrac{(0.58)^3}{7.78} = 0.660$

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19. $W = 1/2(A_{m2} + A_b) \times S_a$

$W = 1/2(0.316 + 4.3968) \times 15{,}000 = 35{,}346$ lbs.

20. $H = 0.785\ P_e(G_m)^2$

$H = 0.785 \times 15 \times 37.745^2 = 15{,}780$ lbs.

21. $H_D = 0.785\ P_e\ (B)^2$

$H_D = 0.785 \times 15 \times 36^2 = 15{,}260$ lbs.

22. $H_T = H - H_D$

$H_T = 520$ lbs.

23. $R = 1/2(C - B) - g_1$

$R = 1/2(40.125 - 36.000) - 0.605 = 1.458$ in.

24. $h_D = R + 0.5\ g_1$

$h_D = 1.458 + 0.5 \times 0.605 = 1.761$ in.

25. $h_G = 1/2(C - G_m)$

$h_G = 1/2(40.125 - 37.745) = 1.19$ in.

26. $h_T = 1/2(R + g_1 + h_G)$

$h_T = 1/2(1.458 + 0.605 + 1.19) = 1.626$ in.

27. From paragraph UA-55 of the Code, for operating conditions:

$M_o = H_D\ (h_D - h_G) + H_T\ (h_T - h_G)$

$M_o = 15{,}260(1.761 - 1.19) + 520(1.626 - 1.19) = 8{,}940$ in.-lbs.

28. From paragraph UA-55 of the Code, for gasket seating:

$M_o = W\ h_G$

$M_o = 35{,}346 \times 1.19 = 42{,}062$ in.-lbs.

29. Gasket seating determines the flange design.

Use, $M_o = 42,062$ in.-lbs.

30. $S_H = \dfrac{f \; M_o}{L \; B \; (g_1)^2}$

$S_H = \dfrac{4.35 \times 42,062}{0.66 \times 36 \times 0.605^2} = 21,000$ psi

31. $S_R = \dfrac{(1.33 \; t \; e + 1) \; M_o}{L \; B \; (t)^2}$

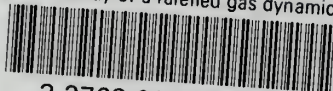$S_R = \dfrac{(1.33 \times 0.58 \times 0.3 + 1) \; 42,062}{0.66 \times 36 \times 0.58^2} = 6,500$ psi

32. $S_T = \dfrac{Y \; M_o}{B \; (t)^2} - Z \; S_R$

$S_T = \dfrac{15 \times 42,062}{36 \times 0.58^2} - 8 \times 6,500 = 200$ psi

33. The requirements for strength are found in paragraph UA-52 of the Code, and are listed in Part A, Section 5 of this thesis.